Heating · Piping and Air Conditioning

Vol. 10, *No.* 6 • *June,* 1938

District Steam Sales Rise, Cost Falls

Trends in District Heating During Last Ten Years

By G. D. Winans*

It is appropriate that attention be given to general trends in district heating this month, for problems of the industry will be discussed in detail at the annual convention of the National District Heating Association meeting in Baltimore June 28 to July 1. . . . Mr. Winans shows what the district heating companies are doing in the field of customers' relations, and toward reduction of capital and operating costs. Efforts in the latter field include the use of compressors to supply comparatively small amounts of high pressure steam, and adoption of improved heat cycles for greater efficiency

WHILE district heating has experienced no spectacular growth during the last 10 years it has withstood the pressure of competition remarkably well during the depression. Fig. 1 shows the total annual sales of 25 companies as reported to the National District Heating Association for a period of nine years.

The cost of the service to the consumer in most cases has followed a downward trend during these years, as shown by Fig. 2. The average net rate of 16 companies has decreased from 89.6 per 1000 lb in 1928 to 78.2c in 1936.

During this period there have been many interesting developments in the industry. These may be classified roughly as those directly affecting relations with customers and as efforts to reduce capital and operating costs.

Improved Customers' Relations

District heating companies have long recognized that good customer relations are particularly important. They have used numerous methods, therefore, of providing better service in order to increase the satisfaction of their customers.

There has been an increased appreciation of the service by building operators, due not only to the decreased cost of steam but also to the increased labor costs of operating small, individually owned boiler plants and to the service rendered by district heating companies in general. Also, the space made available by the abandonment of isolated boiler plants has provided needed space for expansion.

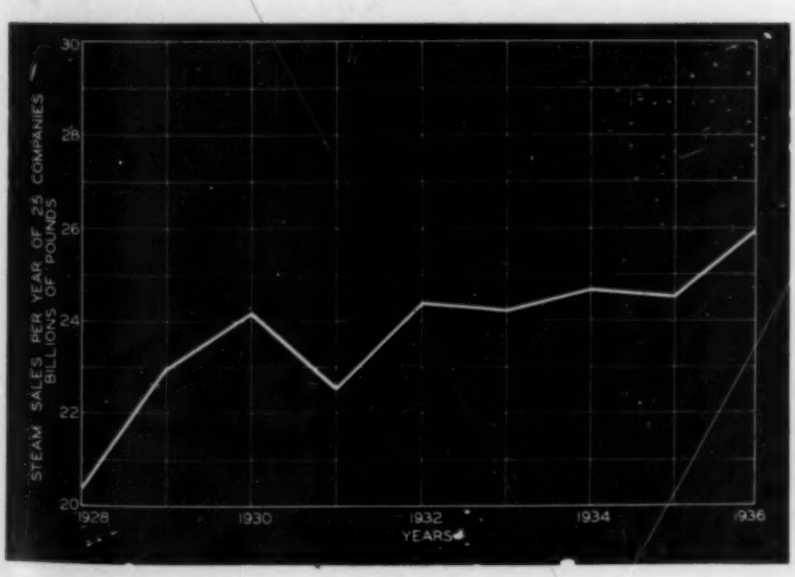
Heating System Controls

The improvement of building heating system controls, most of which has taken place during the past 15 years, has greatly helped the operating companies in providing better service at a lower cost. The building owners have been slow to accept these controls, which vary from simple single units to the very complete installations which are applied to multiple zoned heating systems. They are now, however, showing a greater interest in heating system controls.

The experience of district heating companies has been that, almost without exception, properly installed building temperature control systems not only reduce the heating costs but at the same time give decidedly better heating conditions. This is because the wide temperature variations which accompany hand control are prevented. Due to a desire to give better service at a lower cost, many heating companies have made control installations on a rental basis. This has proved to be good practice

*Engineer of Steam Distribution, The Detroit Edison Co.; Second Vice-President, National District Heating Association.

Fig. 1—Total annual steam sales of 25 district heating companies



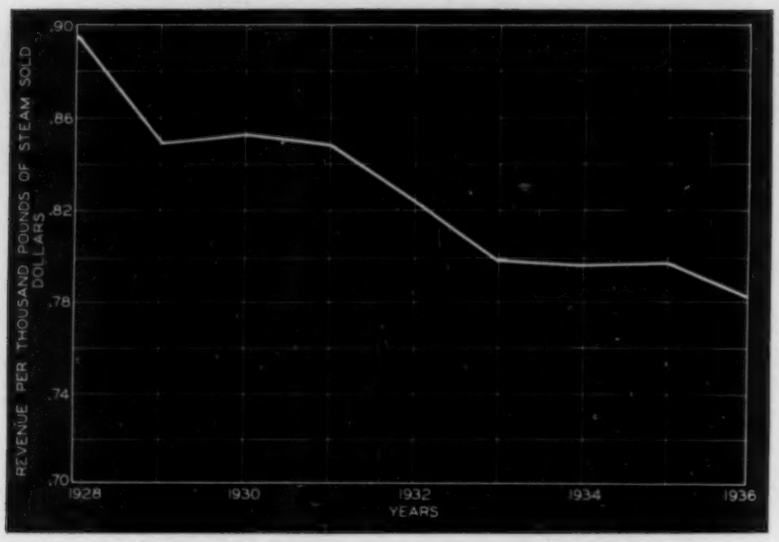


Fig. 2—Revenue per 1000 lb of steam sold by 16 companies

in spite of the fact that these control systems materially reduce the steam sales of the operating companies.

Many operating companies are working in several other ways to reduce the customers' heating costs. The distribution of a booklet entitled "Principles of Economical Heating", published by the commercial relations committee of the National District Heating Association in collaboration with the National Association of Building Owners and Managers, is typical of these practices. This booklet is in its second edition.

Some companies, upon request, make inspections of customers' heating systems and give advice regarding necessary repairs and desirable piping changes which would reduce heating costs.

Flow Meters

Both steam flow meters and condensation meters are used by district heating companies to meter the service. Since condensation meters are dependent to a certain extent upon the condition of the customers' equipment, there has been a tendency to use steam flow meters. This has brought about a feeling of a need for a flow meter better adapted to general district heating metering.

Reduction of Capital and Operating Costs

In the effort to improve their own operating conditions, heating companies have continuously studied means of increasing their output, improving their load factor, and decreasing the cost of the service. Through their interchange of ideas and methods, they have made many advances in this respect.

Air Conditioning

The increased demand for air conditioning seems to offer the greatest opportunity for increasing the steam output. With new developments and methods now available, it has become feasible to use low pressure steam as a means of dehumidifying air for air conditioning purposes. The additional load which this use of steam may furnish will be a most welcome offpeak output to the district heating companies.

Steam Compressors

The use of compressors to supply comparatively small amounts of high pressure steam is of particular importance in those cities where only low pressure steam is available. Steam for use in laundry mangles, pressing machines, and other processes requiring 85 to 100 lb gage pressure is made available from low pressure distribution systems. The use of steam compressors makes it possible to furnish all the steam requirements of some buildings which formerly had small, separately fired boilers, and enlarges the district heating field because new customers who require high pressure steam can be more easily acquired.

There are now six steam compressor installations in Detroit. The first of these was placed in service in 1927 and has given very satisfactory service from an operating and maintenance standpoint. The others have been installed more recently and are giving equal satisfaction.

Plans are now being made to install at least two more steam compressors in a large hotel to supply high pressure steam for the laundry.

Distribution System

The problem of distributing steam economically is being given much attention at present. The cost of underground construction is high and has long been a handicap to the widespread expansion of distribution systems. An earnest attempt has been made to reduce this cost.

[Concluded on p. 371]

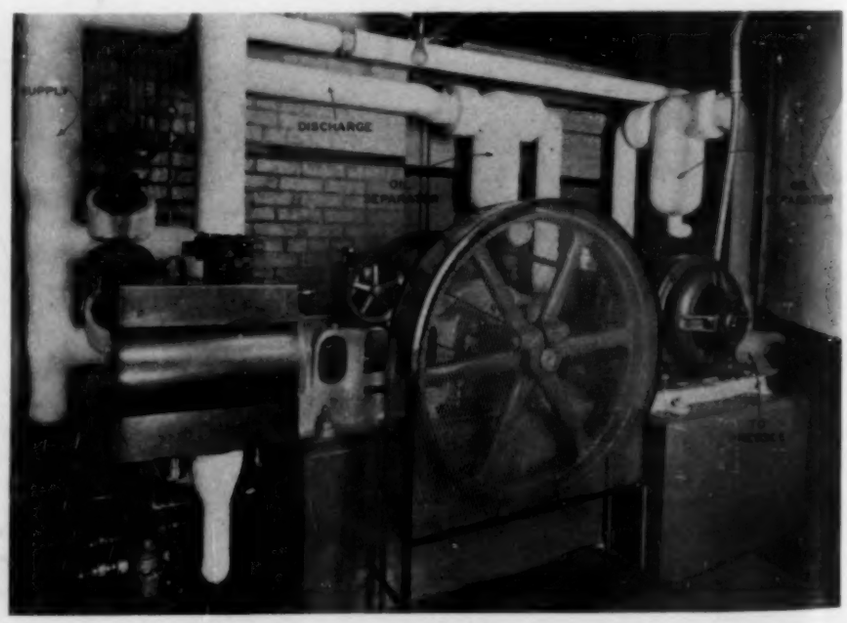


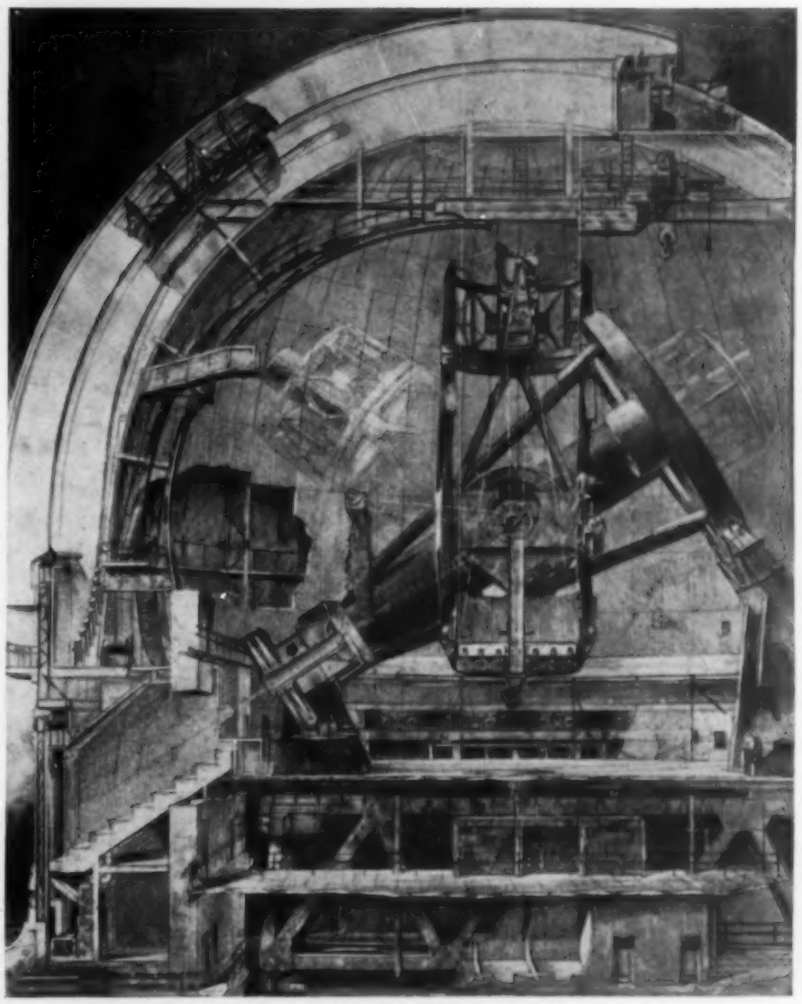
Fig. 3—Steam compressor installation. Receiver in back of compressor cannot be seen

Dome for World's Largest Telescope Presents Unusual Insulation Problem

ASTRONOMERS will be able to study a volume of space approximately eight times greater than ever before when the 200 in. telescope of the Mt. Palomar astrophysical observatory of California Institute of Technology is ready to operate, probably in 1940. With the mighty light-gathering properties of the world's largest telescope, scientists may see 1,000,000,000 light years' into space (twice as far as now), will be able to test the theory of the expanding universe by measuring the speed of nebulae more distant than any yet observed, may learn new facts about the constitution of matter within the stars and their atmospheres.

Many branches of science, engineering and industry are playing parts in the design and erection of this tremendous instrument, initiated in 1927 by a $6,000,000 gift of the Rockefeller Foundation, to increase man's knowledge. Some five years were spent studying possible sites in Southern California and Arizona—areas noted for clarity of atmosphere and nominal temperature variations. The site had to be remote from urban centers and the "sky lights" of such settlements, now and in the future. It had to be accessible by a road with easy grades and large radius curves to permit transportation of the tube and mounting and the heavy parts for the dome construction.

One of the most difficult problems was to procure a suitable 200 in. disk to back the thin metallic reflecting surface of a gigantic parabolic mirror, accurate within two-millionths of an inch at every point of its surface. Two were cast of low expansion borosilicate glass, following months of experimental work, and the best of the two will be used. The second was lifted from its annealing oven in December, 1935, after having cooled for 12 months from its pouring temperature of 2800 degrees. Weighing 20 tons, it was the largest solid piece of glass ever cast. It is now being made ready for the reflecting surface by "figuring" it—giving it the figure of a paraboloid or slightly dished shape so that it will gather light and focus it at a point to be picked up by an eyepiece or split up by a spec-



Cross-section of the dome building and the world's largest telescope, which will vastly increase man's knowledge of the universe

By Channing Turner*

At Mt. Palomar, California (about 60 miles from San Diego) the world's largest astrophysical observatory is nearing completion, will be ready to operate in 1940. . . . . One of the many problems to be solved for this tremendous project—in which many branches of science, engineering, and industry have cooperated—was the method of insulating the dome. A fairly uniform temperature level during each 24 hr period throughout the year is essential in order to prevent movement of the telescope supporting structure and displacement of the mirrors, as otherwise precious "seeing time" would be lost. . . . . How the dome is being insulated is described here, with a brief discussion of some of its features

*Chief Engineer, Alfol Insulation Co., Inc.

troscope. Aluminum approximately 1/250,000 of an inch thick will be deposited on it by a new process to create a reflecting mirror. This process consists of condensing aluminum vapor on the surface under an absolute pressure of approximately 10^{-5} mm of mercury.

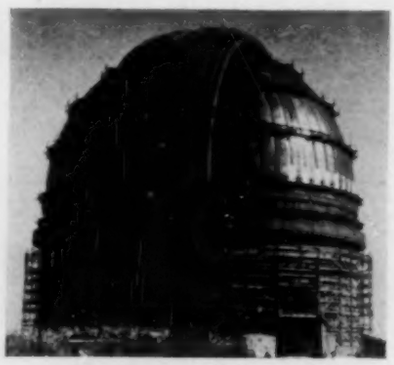
In the November, 1933, HEATING, PIPING AND AIR CONDITIONING, M. M. Lawler described the function air conditioning is performing in the grinding and polishing of the mirror. By providing controlled temperature and humidity conditions for the telescope mirror shop on the campus of California Tech, it has been estimated that the nine years which might otherwise be required for the work would be reduced to three. A controlled temperature is essential to prevent distortion of the glass and consequent inaccurate grinding; controlled humidity is required because the abrasive materials or rouge used for the grinding and polishing are mixed with water, and the consistency of the rouge would vary, because of the evaporation of the water, with the atmospheric conditions in the room.

The contribution of heat engineering to this project does not end with the control of temperature and humidity of the mirror shop, however. The insulation of the dome for the telescope offers unusual problems, for it is essential to maintain a fairly uniform temperature level during each 24 hr period throughout the year (the level depending upon the probable evening temperature). Wide temperature variations would cause excessive expansion and contraction changes in the mounting and frame that support the mirrors in precise relationship, resulting in loss of irreplaceable "seeing time." The plan is to keep the temperature variation in the dome within 5 deg, and the temperature around the 200 in. reflector within approximately 1 deg.

The Dome Construction

The dome, 135 ft in diameter, will be rotated so that the opening, 30 ft wide and over 100 ft in length, is held in line with the telescope wherever it may be pointed. Huge sliding shutters will close the opening all day long and will be opened at night. The area of the opening is about 1/12 the dome area.

From sunset to dawn the dome will be open to the night air. The shutters will be closed at dawn when the atmosphere is at its coldest, and it is necessary to maintain this low temperature inside the dome all day long so that in the late afternoon, when the shutters are opened, the air temperatures inside and outside will be about the same. Otherwise there would be an inrush of cold air into the dome, with chilling and contracting of the steelwork and the rising of warm currents of air, which conditions would interfere with the use of the telescope for several hours, until temperatures had become equalized and stable.



Construction view of dome with shutters partly closed

In summer, the surface temperature of the dome exposed to the sun's heat may be over 120 F. The maximum air temperature may be 90 F or so, and the minimum temperature at night might be about 70 F or may be as low as 60 F. In summer it is then desirable to maintain the temperature inside the dome on some days from 70 to 75 F and on other days the range may be lower, as from 60 to 65 F. The problem is to maintain a fairly uniform temperature level throughout a 24 hr period, the exact level depending upon the probable evening temperature. On cold winter days the maximum air temperature may be 30 F and the minimum 15 F, so in this season the desirable inside temperature will be considerably lower than in summer.

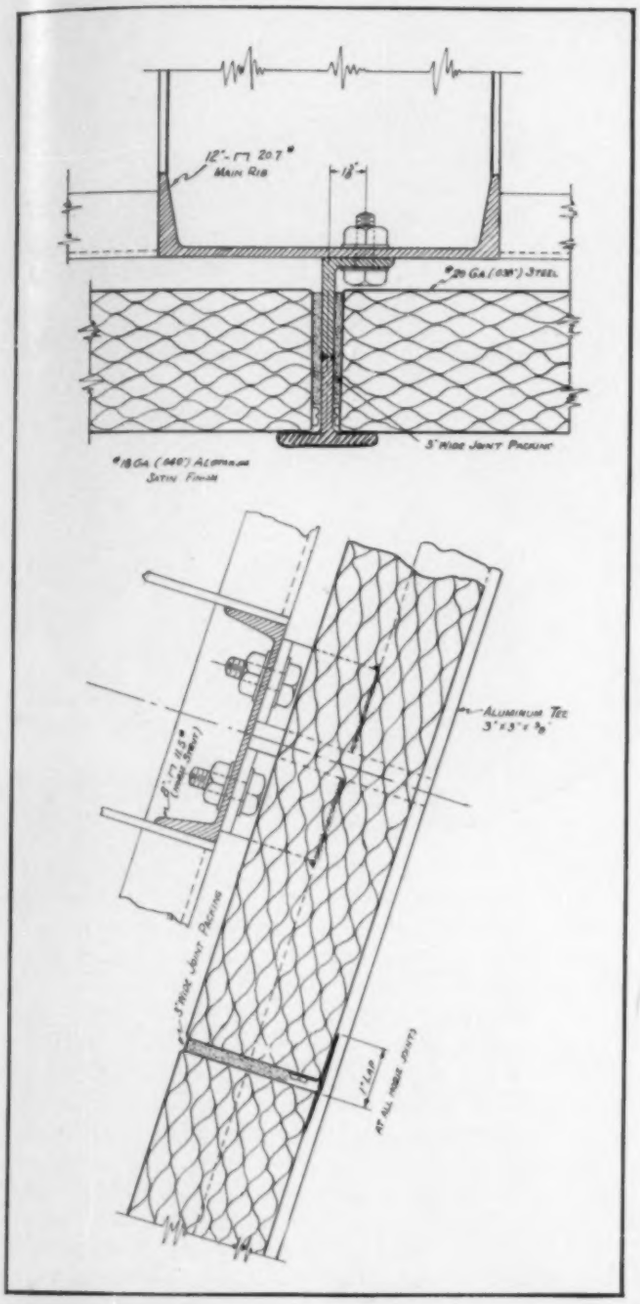
The most satisfactory way to control the situation and maintain night temperatures throughout the day is to use insulation to keep the heat from penetrating the dome. In addition to high thermal efficiency, the insulation of the vast inside area of this revolving dome must be unaffected by possible vibration; must be fireproof; must be held firmly in place; must not sag or warp; and must be enduring. It was also deemed desirable to choose a type which would provide an attractive architectural effect. Aluminum foil was selected.

Method of Insulating

The aluminum foil is made up in metal panels 5 in. thick holding twelve layers of crumpled foil. Layers are separated by contact between high points and completely fill the depth of the panel. Referring to the detail drawings, the dome is built up of steel plates butt welded together and supported on I beams which are curved to the radius of the dome and erected in vertical planes. These I beams are supported by horizontal trusses which, in turn, are carried by larger trusses arching up in circular arcs from the springline of the dome to the large box girders of semicircular form which frame the opening of the dome. The arching trusses are 3½ ft in depth and the insulating panels are placed in line with the inside faces of these trusses, to form a ceiling with concave spherical surface. This spherical surface is 3½ ft from the inside surface of the dome plates, except at the very top of the dome, where this insulating ceiling is flat and passes horizontally just under the box girders and extends to the sides of the opening.

The insulating panels are supported and held in line by structural T's of aluminum standing in vertical planes and rolled to conform to the curvature of the dome. Each panel fits between the stems of the T's and is retained by the flange on each side. The panels are clamped to the T's by wedges and plates. This construction holds the panels firmly in place and allows them to be erected with ease.

The side panels are about 7 ft high and decrease in width from about 4 ft at the springline to about 2 ft at the flat ceiling. They are curved inward to conform with the shape of the dome. The twelve layers of foil are held in pan shaped trays of light sheet steel covered with a sheet of aluminum, rubbed to a satin finish.



***Above*—Cross-section showing method of attaching aluminum tees to bottom of main ribs of dome. *Below*—Section showing attaching of aluminum tees to bottom of horizontal struts**

This concave aluminum surface forms the interior surface of the dome and this attractive metal finish is set off by the slender lines of the T's which rise from the observation gallery and arch high overhead, converging toward the top of the dome. This construction not only provides an interior sheathing of attractive appearance, but has a high utility value also, because the metallic surface of the aluminum sheets has high reflectivity and low emissivity of radiant heat. This means that the transfer of radiant heat between the telescope and the revolving dome will be reduced to a minimum, which will add materially to the insulation of the telescope.

In addition to the foil insulation applied in the panel form in the dome, loose crumpled foil has been used to insulate outside walls and inside walls and ceilings of the three stories inside the concrete dome building, as well as for insulation of the shutter and iris diaphragm of the 200 in. reflector.

Lower Walls Are Double

The lower walls of the building—the stationary part below the revolving dome—are double, with a 12 in. ventilated air space between an outer 1½ in. gunite wall and an inner insulated wall. This design was developed to keep the lower part of the dome construction slightly cooler than the upper part and to permit a rapid dissipation of heat from the outer wall, so as not to cause local disturbance and interference with "seeing time" in the early evening. Several workrooms in the stationary part of the dome building, which extends approximately 30 ft from the ground, are to be heated and air conditioned.

The height of the entire building, including the rotating dome, is about 135 ft from the ground to the top of the dome area.

District Steam—

[Concluded from p. 368]

Several new types of steam main conduits have been developed with this in mind. The improvements in welding technique and in methods of testing welds have brought about universal acceptance of welding piping for both underground and plant construction. Both gas and arc welding are used.

Steam Generation

Few additions have been made to steam generating equipment recently, although many improvements have been made in existing installations. For the most part, heating plant development runs hand in hand with the development in electric power plants.

When additions or new plants are needed, new heat cycles will no doubt be applied to take advantage of the greater efficiencies to be obtained by the use of higher temperatures and pressures. Some of the cycles are described in detail in a paper, "A Comparative Study of By-Product Generating Cycles", by M. W. Benjamin and R. G. Felger, N.D.H.A. "Proceedings," 1933.

One installation using an improved cycle is being installed this year. This will consist, in part, of two evaporators, each having a net capacity of 100,000 lb per hr of steam. These evaporators will be supplied with steam from a back pressure turbine. The steam generated in the evaporators will be used to supply the steam distribution system furnishing steam to the downtown district. Others will follow as soon as there is sufficient demand for more plant capacity.

Advisory Council

Last June, President T. E. Purcell of the National District Heating Association appointed an advisory council. This council has, as the basis of its organization, the development of district heating on sound economic and engineering lines. The council is made up of nine of the older members of the association, and should be able to accomplish much toward the advancement of district heating.

After College—— Air Conditioning?

By Herbert Raisler

THE engineering graduate is faced not only with the problem of getting a job, but of getting one in the particular field best suited to his abilities. In recent years, students have leaned heavily toward air conditioning. In many universities where air conditioning courses are offered, almost 30 per cent of the eligible mechanical engineers are enrolled in them. This interest can be traced largely to the impression that air conditioning is new, and consequently should offer great opportunities to young men.

Realizing this interest, information from several companies in the industry and from some of the foremost engineering schools has been assembled to present a picture of the opportunities for engineering graduates in air conditioning. The study is based upon data and comments obtained from many sources, and does not reflect the attitude or procedure of any one organization.

Scholastic Preparation

In a subject as fast-changing as this, it is difficult for a university to maintain courses which will immediately prepare its students for a useful position. Sections of textbooks become outdated before publication has been completed, and most men who are not active participants in the business of the industry find it impossible to keep fully abreast of the times.

This has limited the academic preparation obtainable. Most technical schools offer one or two courses dealing with the subject. Some have adopted a system whereby it is possible for its students to major in this work in the senior year; others allow it as a thesis topic. Undergraduate work in air conditioning is of course of value; for the most part, however, these courses deal only with the basic theory, and do not incorporate practical applications.

Because of this, most of the companies engaged in the industry have taken it upon themselves to teach the subject and prepare men for useful positions in their organizations. With this system, anyone who has graduated from an engineering school is equipped to start with a company in the air conditioning business.

Obtaining Employment

The methods by which a position in the air conditioning industry may be obtained are varied, and depend upon the type of work and the company involved.

Most firms have a definite policy that is followed in selecting men. Each year (depending upon business conditions) a certain number of students are carefully chosen from the graduating classes of different colleges. These are usually selected in one of two ways. The company either sends its personnel director to interview students at the various universities, or arranges for interviews at its offices. The latter come about as a result of an inquiry from a prospective employee, or because of some qualified person's recommendation. In either case, an application is filed and the man is interviewed by one or more members of the company.

Some of the companies operate so as to train men for their branch offices or distributors. These men are usually selected from the particular section of the country where the office is located, and in almost all cases the branch or distributor managers have the final say in determining who shall become a member of the local organization.

New men are divided into two distinct classifications; (1) those who are to become sales engineers and work in the merchandising and business end of the company, and (2) those who are to become research or production engineers and work in the development departments. Usually, this division is kept in mind from the beginning, and sales personalities are segregated from those whose interests are strictly technical.

Another method of entering the industry is through companies indirectly related to it. Some firms treat air conditioning as an auxiliary part of their main line. Others are allied to the field in that they manufacture companion products. Employment in these organizations necessitates undertaking a certain amount of pre-

liminary training of a general nature, but usually, if conditions allow, such companies encourage those whose interests lie in a particular branch of the business to enter that branch.

Employment opportunities, then, may be classified as to type of company and type of position. The field is broad and its scope unlimited, but like any other industry, selections are carefully made, and a good technical background is an ever important requisite.

Training in Industry

The lengths to which companies go in training their engineers is quite unusual. Almost all organizations have a definite routine which the student engineer follows. The duration and nature of this training depend upon the company, type of job, individual, and general business conditions.

The graduate first spends a short period familiarizing himself with the products, methods, and physical workings of the organization. During this orientation period, a certain amount of shop work and plant touring is performed; the time devoted to this may vary from two weeks to six months. This period is sometimes extended to allow time in the research laboratory where testing and development is encountered. If a man is being trained for purely technical work, this department will be greatly stressed and a year or more may be spent learning intricacies of laboratory technique.

For the sales engineer, a far more important phase of the training takes place in the layout department, where calculations for actual jobs are made. Here, the equipment that the company deals in must be employed

---

Of recent years, engineering graduates have leaned heavily toward air conditioning, feeling that in such a fast moving field there must be many opportunities for young men. This article is intended to help the graduate in his decision, aid H. P. & A. C. readers in advising younger men when their opinions are asked .... The author—one year out of school—has worked for one of the major heating and air conditioning manufacturers and has been a member of the company's student training course. He previously had two summers' experience with two different companies closely allied to the industry, and has interviewed by mail and in person several other manufacturers and universities for the information he summarizes .... His purpose is to describe what the college graduate may expect if he decides to cast his lot with the air conditioning manufacturer

---

to satisfy the specifications of owners, engineers, and architects. Each job is different, and the layout work comprises one of the finest types of experience obtainable.

Student engineers who are to be placed in branch offices are next sent out to obtain field experience. There, the men follow a procedure similar to that of the main office, only in this case the work is more directly applicable to the particular branch office under consideration.

In each instance, the men are under the complete supervision of the company's engineers and executives. They are at liberty to discuss any problems with them, and in that way manage to obtain a broader understanding of the subject.

The practical work is supplemented by lectures, home problems, outside reading, and examinations which ordinarily take place after hours. Lectures are given one or two nights a week either by a series of the company's engineers, or an instructor delegated to perform that part of the training.

The duration of the entire educational period may vary from six months to five years or more, but more commonly, a year is required thoroughly to train a college graduate. After that, he is considered to have passed the stage of learning and entered that of useful work. From general inquiry, 75 to 80 per cent of the men trained remain with the company, and in many cases the more important technical and executive positions in the organizations are now occupied by men who have been trained in this manner.

Education in industry is not an innovation, but the extent to which it has been undertaken in the air conditioning field is great.

Opportunities Do Exist

Today's comfort conditioning is a far cry from the installations of 20 and 30 years ago. Great progress has been made, and public interest has been seriously aroused, but it is well to remember that air conditioning is no more than a modified combination of three older fields: heating, ventilation, and refrigeration. Its principles hold no mystery, and its applications are not far removed from these component parts.

Opportunities in air conditioning do exist, but contrary to many exaggerated statements, the industry is not an economic cure-all for financial ills. The field is new, and adjustments are continually being made. These adjustments many times result in excellent opportunities for comparatively young men.

A good future exists for capable engineers—providing a sound technical knowledge, founded upon academic training and industrial experience, is accompanied by a sincere desire to help contribute to furthering the field. Those who have pinned their hopes on entering an industry wherein ability is lacking and new blood is sorely needed will be disappointed.

The industry is well equipped educationally, and the possibility of obtaining worthwhile experience is evidenced by the many ways that training is undertaken. The companies comprising the major part of this business employ great numbers of college graduates yearly, recruiting them from universities throughout the country. Their success and the future of the industry depend upon their ability.

Air Conditioning Mines

The Bureau of Mines recently issued Information Circular No. I. C. 7001 on "Review of Literature on Conditioning Air for Advancement of Health and Safety in Mines," by D. Harrington and Sara J. Davenport. Among the phases of the subject discussed are general foulness in underground air, the gases encountered in mines, and dusts and the prevention and treatment of dust diseases. A bibliography of 70 references is included.

Valves

Table 1—Classification of Valves

	Class 1	Class 2	Class 3
Class A	Swing check valves	Angle valves Y valves Lift check valves, angle type	Globe valves Lift check valves globe type
Class B	Wedge gate valves D. D. taper seat gates	Needlepoint angle valves Wedge gates—venturi type D. D. type seat gates—venturi type	Needlepoint globe valves
Class C	Cocks Lubricated plug valves, full area D. D. parallel seat gate valves	Cocks—venturi type Lubricated plug valves—reduced area D. D. parallel seat gates, venturi type	

—a Review of Available Types and Their Features

By F. H. Morehead*

A GENERAL classification and discussion of types of valves was given in December.[1] The types were classified according to the path of the fluid through the wide open valve, and the nature of the opening and closing mechanisms. In Table 1, they are arranged from left to right in order of their tendency toward obstruction of flow, and from top to bottom in the order of their tendency toward wearing, scoring, galling and seizing.

Special Features of Valves Illustrated

The following illustrations will serve to visualize the differences in construction of the commonly used valves outlined in Table 1, and at the same time will illustrate various special design features which characterize the different lines of valves. The illustrations are intended to represent types rather than particular valves.

Fig. 1 shows a typical design standard bronze globe valve, Class 3A, rated for 125 lb maximum steam working pressure. Competition demands that these valves shall be sold at a low price and they consequently do not have some of the features which are incorporated in the higher priced valves. As usually made, they have angular seats, valve stem threads exposed to the action of the pipe line fluid, male thread on the bonnet screwed into female thread in the body, and stuffing box without gland follower.

For severe service conditions a stuffing box having a gland follower may be preferred, because it keeps the packing away from the threads of the stuffing box nut and provides a smooth walled container in which the packing can be uniformly compressed without injuring it. It does not seem possible, however, to standardize the opinions and preferences of all valve users as is evidenced by the fact that some specify stuffing boxes without gland followers even on the higher priced valves, on the theory that with this construction the stuffing box contains more packing and consequently will not have to be repacked as frequently. This of course may be true when a stuffing box without gland is compared with a gland type stuffing box of inadequate design, but when the proportions are ample the superiority of the gland type stuffing box hardly seems open to question, in the writer's opinion.

A complete line of globe valves for a given pressure is usually accompanied by the following companion lines: (1) Angle valves. (2) Lift type globe check valves. (3) Lift type angle check valves.

Cross valves having three pipe connections are also used to some extent, but the demand for them is comparatively small and they are not usually available for shipment from stock. They are usually made from the angle valve patterns having two pipe connections above the valve seat, and one below, but they may also be made from the globe valve patterns, in which case two pipe connections are below the valve seat and one above.

Fig. 2 shows a modified form of globe valve called a needlepoint valve, Class 3B, which is designed especially for close regulation of flow of liquids, vapors and gases. It will be noted that the disc is made integral with the stem. This causes some sliding of the seats across each other at the instant of closure and places the valve in Class B instead of Class A. In the usual globe valve construction the disc is connected loosely to the stem and does not rotate with the stem after contacting with the body seat. The long taper seat aids in securing a tight closure, and coupled with the fine pitched stem thread makes possible close adjustments of the flow through the valve. The port opening through the seat of the valve is not full pipe size because closer regulation is secured with a reduced port opening, and valves of this type are often made with two or three different sizes of seat openings in order to accommodate the range of conditions encountered in service. For instance, a ½ in. needlepoint valve for moderately heavy fuel oil would probably have a seat

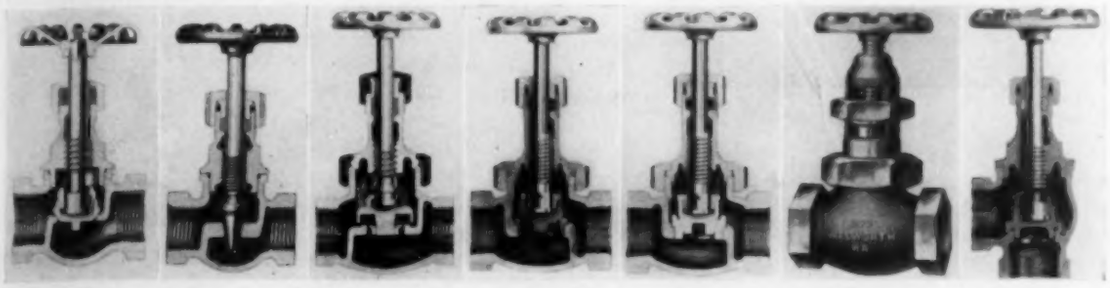
The vital part that valves play in modern piping practice is exemplified by the many kinds of valves which are used for controlling and regulating "industry's most important transportation system" . . . The author has attempted in this review to present a general picture of the basic types of valves, and to indicate, insofar as is possible, the particular applications, inherent advantages, and limitations of each

*Vice-President in Charge of Engineering, Walworth Co.

[1] See Heating, Piping and Air Conditioning, December, 1937, pp. 734-736.

port about $\frac{5}{8}$ in. in diameter, while the same size valve used on gasoline or on compressed air would probably have a seat port about $\frac{1}{8}$ in. in diameter.

Fig. 3 shows a bronze globe valve with a flat seat and renewable composition disc. This valve, Class 3A, has a number of attractive features. Globe and angle valves having metallic seats and discs are seldom made with flat seats because considerably more valve stem thrust is required to close the valve tight against the line pressure than for an angular seat valve. The reason for this is that with flat seat valves there is no wedge action when the disc is forced against the seat. Consequently, the stem thrust must be greater in order to secure the same unit pressure between the disc and seat. Also, the seat contact extends farther outside of the port and the stem thrust is usually spread over a larger seat contact area, which further increases the need for additional stem thrust. Flat seat valves with composition discs, however, are readily closed tight because less unit pressure between the disc and seat is required to seal the joint on account of the comparatively soft and yielding nature of the composition disc.



Left to right: **Fig. 1—Standard bronze globe valve, Class 3A . . . Fig. 2—Needlepoint valve, Class 3B . . . Fig. 3—Globe valve with composition disc, Class 3A . . . Fig. 4—Regrinding globe valve with renewable seat and disc, Class 3A . . . Fig. 5—Regrinding globe valve with renewable seat and disc of the plug disc type, Class 3A . . . Fig. 6—Regrinding globe valve with renewable seat and disc, outside screw and yoke type, Class 3A . . . Fig. 7—Bronze angle valve, Class 2A**

A further advantage of the composition disc is that damage to the seats caused by closing the valve on hard particles of foreign matter is largely confined to the renewable composition disc, and even though the metallic body seat should be injured to some extent the softer composition disc would still be able to make a tight closure.

Composition discs, however, are subject to temperature limitations and care should be exercised to see that a disc suitable for the service conditions is employed. Discs intended for steam and other hot service are regularly furnished in valves of this type unless otherwise specified. These discs are relatively hard when cold, but soften to some extent at higher temperatures. Although the hard discs may be used on both hot and cold service, it is preferable on cold service to use soft composition discs which valve manufacturers are prepared to furnish when specified.

Fig. 4 shows a regrinding renewable seat and disc bronze globe valve, Class 3A, rated for 300 lb W.S.P.; it is equally effective for oil, water, air and gas. It is of the union bonnet type, which construction permits regrinding of the seat and disc without removing the valve from the pipe line. The renewable seat and disc are made of various alloys which are resistant to corrosion, erosion and heat. Materials for seats and discs are selected to meet service conditions. In some cases, hard copper-nickel alloys are used. Another combination is hard copper-nickel alloy disc and a stainless steel seat ring. These metals resist galling and wire drawing. The long stem thread provides ample bearing surface to assure long life, and the combination of these features results in a valve particularly suitable for severe service conditions.

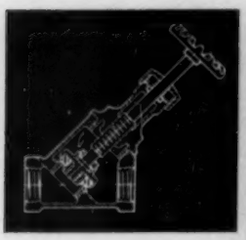
Fig. 5 shows a plug disc bronze globe valve, Class 3A, which is similar to the valve shown in Fig. 4 except that it is equipped with a plug type disc having an extra wide seat contact surface. It will be noted that this valve seat is somewhat similar to that of the needlepoint valve, Fig. 2, except that the disc swivels on the stem and the port opening is not so drastically reduced. Like the needlepoint valve, this valve is intended primarily for throttling service and for regulating flow. Due to their wide contact surface the seats withstand the erosion incident to throttling service. These valves are available with seat and disc materials as described above and also with high carbon stainless steel seats, hardened to 500 Brinell.

Fig. 6 shows an O.S.&Y. type regrinding renewable seat and disc bronze globe valve, Class 3A, similar in all respects to Fig. 3, except that the stem threads are outside the valve body so as to be removed from the deteriorating effects of heat and corrosion and so they can be readily lubricated to prevent excessive wear due to frequent operation of the valve. The yoke also reinforces the stem and protects the stem threads from damage by rough usage. The outside screw and yoke, in addition to the regrinding and renewable seat and disc features, make such a valve desirable for extremely severe service conditions.

Fig. 7 shows a dome blow-off bronze angle valve, Class 2A, which will serve to illustrate the general design of angle valves as compared with globe valves. It will be noted that the flow passage through the valve is more direct than it is through a globe valve, which of course results in less pressure drop. The objectionable pocket formed by a globe valve diaphragm is also missing. Angle valves, therefore, should be used in preference to globes whenever the piping layout will permit their use. Companion lines of angle valves are available having special design features similar to the globe valves previously illustrated.

A special feature of the valve shown in this illustration is the disc and seat construction which is designed for blow-off service, skimming lines, and similar conditions, wherever a fluid containing grit or dirt is blown off under pressure. The external type angular valve seat, being of considerably larger diameter than

the port, provides increased flow area which reduces the velocity across the seating surfaces and the lip under the disc enters the port before the seats come in contact, thus further reducing the velocity and tendency toward erosion in the critical position just before closure is effected. The design of the seat also tends to prevent foreign matter from collecting and interfering



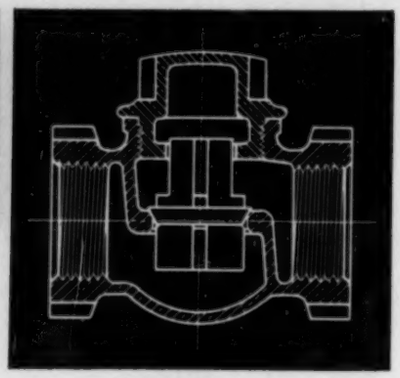


Fig. 8—Bronze Y valve, Class 2A . . . Fig. 9—Lift check valve, globe type, Class 3A

with the perfect seating of the valve. To prevent unnecessary erosion of the seat and disc the valve should be opened wide when blowing off.

Fig. 8 shows a composition disc bronze Y valve, Class 2A, rated for 150 lb W.S.P. Valves of this type can be installed in a straight run of pipe and provide a more direct flow passage than globe valves. Their construction features are similar to those of globe valves, except that the angular position of the bonnet tends to increase the length, weight and cost of the valve.

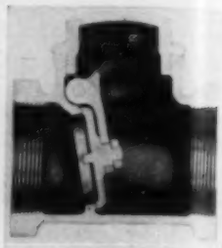
Fig. 9 illustrates typical construction of a globe type lift check valve, Class 3A. The travel of the disc is directed by guides located above and below the seat so that accurate seating is secured. This type of valve, however, in addition to introducing considerable obstruction to flow, has the disadvantage that it can be used only in horizontal pipe lines. Angle check valves of the lift type are constructed in a similar way, and lift check valves for use in vertical pipes are also made and used to a limited extent.

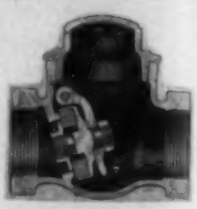
Fig. 10 shows a bronze swing check valve, Class 1A, rated for 125 lb W.S.P., and illustrates typical construction for this class of valve. Flat seats are necessarily used because the hinge pin construction does not afford sufficiently accurate guiding to insure proper seating of an angular contact face. Valves of this type are usually furnished with metal to metal seats, but for cold water service leather faced discs are often used in order to secure tighter seating. Swing check valves are used extensively because they offer little resistance to flow and are adaptable for use in both horizontal and vertical pipe lines.

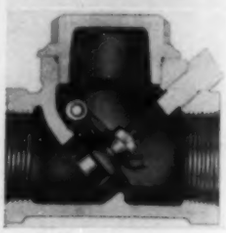
Fig. 11 shows a composition disc bronze swing check valve, Class 1A, rated for 125 lb W.S.P. As in the case of other composition disc valves, hard discs are used for hot service and soft discs are furnished for cold service when so ordered. The advantages of the composition disc are that it provides a tighter seat, is readily renewable, and protects the metal body seat by taking most of the wear. They cannot be used successfully, however, with pipe line fluids that deteriorate the disc too rapidly.

Fig. 12 shows a regrinding extra heavy bronze swing check valve, Class 1A, rated for 300 lb W.S.P. Its construction permits regrinding of the seat contact faces without removing the valve from the pipe line, so that the valve seats can be repaired when they become injured, thus materially increasing the useful life of the valve.

Fig. 13 shows a standard bronze wedge gate valve, Class 1B, rated for 125 lb W.S.P. Gate valves are of three general types, namely, solid wedge, double disc parallel seat, and double disc taper seat. Each of these types has certain inherent advantages and disadvantages, while all three types have the common advantage of straight way flow. The outstanding advantage of the solid wedge gate valve which accounts for its extensive use is its simplicity in







***Top to bottom:* Fig. 10—Swing check valve, Class 1A . . . Fig. 11—Swing check valve with composition disc, Class 1A . . . Fig. 12—Regrinding swing check valve, Class 1A**

design. It has only four essential parts; namely, the body and bonnet, for confining the fluid; the disc for controlling flow; and the stem for raising and lowering

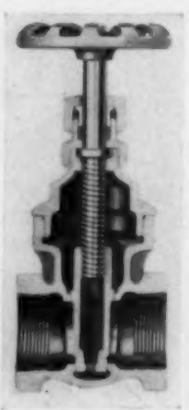


Fig. 13—Wedge gate valve, Class 1B . . . Fig. 14—Wedge gate valve, outside screw and yoke type, Class 1B

the disc. This type of valve is compact and rugged, has the least possible number of wearing parts, and is therefore well adapted for withstanding corrosion, incrustation and hard usage. The gate is guided in the body and the seats are inclined at a total included angle of approximately 10 deg. Consequently, the gate and body seat faces do not slide against each other under pressure until the valve is nearly closed. This reduces the tendency toward scoring and galling, but at the same time increases the tendency toward erosion and wire drawing as compared with the double disc parallel seat gate valve.

The solid wedge gate valve also has two more or less important disadvantages, namely, inflexibility and difficulty in repairing damage to the seats. In manufacturing the solid wedge gate valve, the tapered wedge must be made to fit the tapered body

seats very accurately or the valve will not be tight. Because of the accurate fit required between the body seats and the disc it is difficult to repair the body seats and the disc except in a properly equipped valve factory if, by some mischance or by severe service, they become damaged.

Bronze valves of the type shown in the illustration are comparatively inexpensive, and it usually does not pay to repair the seats, but large iron body and steel gate valves are often repaired several times before being replaced with a new valve. It will be noted that this valve is of the non-rising stem type with the stem threads inside the valve body and exposed to the pipe line fluid. For some service conditions it is desirable to protect the stem threads from heat and corrosion, and in such cases the outside screw and yoke wedge gate valve, shown in Fig. 14, is recommended. This type of valve also has the additional advantage that the length of valve stem projecting above the handwheel indicates at a glance whether the valve is open, partly open, or fully closed.

Fig. 15 shows a standard double disc parallel seat bronze gate valve, Class 1C, rated for 125 lb W.S.P. In valves of this type the downstream disc is pressed against the body seat during closure by the flow in the line, and while the disc and body seats are protected from erosion and wire drawing, the sliding action of the two seats under pressure throughout the full travel of the disc subjects them to the possibility of scoring and galling.

The spreading device seats the discs firmly after they reach the closed position, forcing the upstream disc to its seat against the line pressure and adding to the contact pressure of the downstream seat. The principal advantages of this type of gate valve are flexibility and ease of repairing seats. The two discs are free to adjust themselves to the body seats, and for this reason the angle between the seats and the axis of the valve does not need to be so accurately controlled in order to assure a tight valve. This feature, coupled with the fact that the seat faces are at right angles to the axis of the valve, makes it a comparatively simple matter to reface the body and disc seats in an ordinary lathe. The principal disadvantages of this valve are the greater number of working parts and wearing surfaces, the possibility of the spreading device becoming clogged due to sediment and incrustation, the possibility of premature operation of the spreading device due to striking an obstruction or some other cause, and the possibility of the spreading device not releasing when opening the valve. Difficulty with the spreading device is more likely to occur when the valve is installed in an inverted position or with its stem horizontal.

Fig. 16 shows a double disc taper seat bronze gate valve, Class 1B, rated for 200 lb W.S.P. Valves of this type do not require a spreading device as they close against the inclined body seats like a solid wedge gate valve. The discs are free to adjust themselves to the body seats, and consequently the angle of the seats does not have to be so carefully controlled as in the case of a solid wedge valve. This advantage, however, is accompanied by the disadvantage of an increase in the number of parts and wearing surfaces exposed to the ravages of abrasion, corrosion and incrustation.

Fig. 17 shows a standard square head bronze steam cock, Class 1C, rated for 125 lb W.S.P. This is a conventional type cock and the illustration indicates its simple construction and practically straight way flow. As usually made, cocks have a full size port opening, but the port instead of being round is trapezoidal in shape. Round port cocks are also used to some extent for service conditions where the pipe line fluid builds up solid deposits on the walls of the conduit and has to be intermittently cleaned out. The round port facilitates cleaning by permitting passage through the valve of swabs, scrapers, go-devils, tube cleaners, etc. The round port design, however, necessarily involves the use of a larger plug diameter.

Fig. 18 shows an extra heavy oil country Essex lock pattern iron cock, Class 1C, rated for 350 lb non-shock oil and gas pressures at ordinary air temperature. Cocks of this type are particularly advantageous on loading racks, tank cars, or oil and gas lines from the field where it is desirable to prevent tampering with the lines.

The illustrations used in this article for displaying the features of the various types of valves are largely bronze valves, but iron and steel valves are also made in all the different types and for a great variety of pressure and temperature service conditions. Special valves for particular industries are also used extensively, and valves made of special alloys to withstand unusual service conditions created by new industrial processes are being called for to an increasing extent.

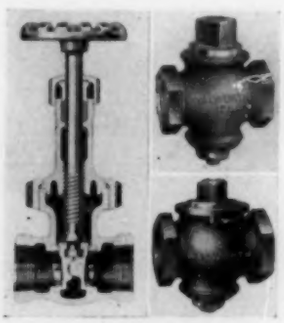


Fig. 15 (*left*)—Double disc, parallel seat gate valve, Class 1C . . . Fig. 16 (*center*)—Double disc, taper seat gate valve, Class 1B . . . Fig. 17 (*right above*)—Bronze steam cock, Class 1C . . . Fig. 18 (*right below*)—Oil country lock pattern iron cock, Class 1C

Power Conference Papers Published

The papers presented at the recent Midwest Power Conference sponsored by Armour Institute of Technology, with the cooperation of other universities and engineering societies, have been published in a 170 page paper bound volume of proceedings, the price of which is $2.00.

Included are discussions on research in the power industry, fuel problems, metallurgy, steam erosion of ferrous and non-ferrous metals, operating problems, heat energy costs, power requirements for residential air conditioning, and others.

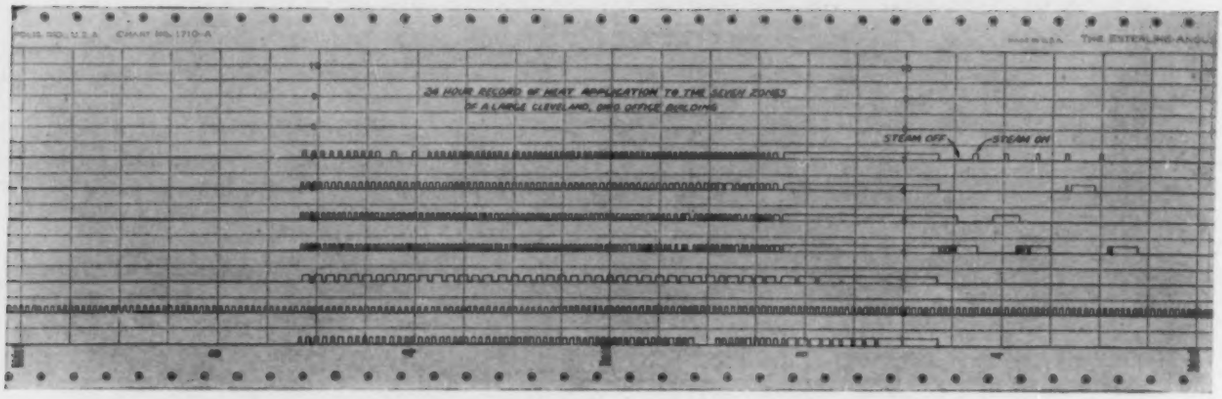


Fig. 1—A 24 hr record of the action of seven valves controlling the heating of seven different zones of a large office building. There can be no question as to how the heating system is handled when records of this kind are made. Note that six of the sections were heated only during the working day, but that Section No. 2 was heated day and night. Note also the slower action of the No. 3 regulator. Janitors and watchmen follow instructions when their actions are recorded

Graphic Records Aid Operation

Graphic Meters Find the Trouble, Show Ways to Lower Costs

GRAPHIC representation consists in arranging on a chart, a curve or series of lines which shows the relationship between values of one or more quantities at different times. A graphic meter is an instrument which makes a continuous measurement of some quantity and plots a record of these measurements on a moving chart, corresponding to time, in the form of a graph. The ways in which graphic meters are used in industry are of almost infinite variety, but in nearly every case the real purpose is to discover just what is taking place in order to lower cost, increase output, improve the process, or better the quality of the product or service.

Graphic electrical meters may often be used to advantage in the operation and maintenance of heating, piping, and air conditioning equipment in plants and buildings. It is the purpose here to show some of the applications of such instruments by citing actual cases.

Supervising Heating Systems

As a rule, the heating schedules for different parts of a large building are not the same. Some sections may be used night and day, others only during the day; some every day in the week, and others irregularly. Furthermore, the different parts of a tall office building will have widely differing exposures to wind and sun. It is the practice to divide the building into zones and to heat each zone automatically on a predetermined schedule.

The automatic control system must of necessity be provided with manual bypass switches to enable an operator to vary the automatic schedule to meet varying requirements of occupancy. Since the performance of the system can be impaired seriously by abusive manipulation of the manual switches, it is desirable that there be a continuous 24 hr record of the system's operation.

By the use of a graphic time recorder, abuses can be detected and corrected. The recorder may even be the main factor in determining whether or not a control installation will be successful. In Fig. 1 is a chart from such a recorder, showing the functioning of the heating system in a large office building comprising seven zones.

Such records have a three-fold value To the owner of the building, they show conclusively at a glance whether or not the control system is functioning. They also show him what time the steam is turned on in the morning and off at night. They show him, too, whenever he has been called upon to supply heat during the evenings or on Sundays for occasional tenants.

To the installer of the control system, the records are invaluable during the period just after the installation is completed and during which he is making adjustments and corrections. They enable him to know definitely whether or not the control has been bypassed.

Feeling a Pump's Pulse

Because of its simple construction, a pump usually makes little disturbance when it gets out of condition. Often pumps are driven long after they should have received the attention of the maintenance department.

The charts in Figs. 2 and 3 were obtained in a plant that manufactures storage batteries. In the systematic surveys of the plant equipment, the chart from the motor driving a lead pump indicated violent fluctuations in the power used, showing that something was wrong inside the pump. It was taken down, the trouble located, and repairs made. When put back in service, operation was normal again, as shown by Fig. 3.

By J. W. Esterline*

Many of the real causes of inefficiency are intangible, unsuspected, and unseen. Most plants and buildings correct visible causes of waste and loss; the other losses persist because they have not been discovered. . . . In many cases, graphic instruments may be used to find ways of improving operation and reducing operating and maintenance costs. Actual examples reported here of cases involving heating, piping, and air conditioning equipment suggest the possibilities

In a large paper mill charts were taken during one of the periodic surveys that this company makes. The first disclosed occasional variations in the motor load, although the pump was operating under a steady load. When the pump was shut down and disassembled, the impeller was found to be loose and wearing on the end of the housing. This was corrected.

Saving Exhaust System Power

Exhaust systems in woodworking, shoe manufacturing, metal polishing and other industries are designed to provide sufficient suction to handle the offal when all machines are operating. Operators are often of the opinion that to close the ducts not in use would cause the power requirements to increase.

An alert engineer, noticing that the exhaust fan in a polishing room ran all the time, regardless of the number of machines being used, decided to find out whether he could reduce the power consumed when some of the machines were idle.

He placed his graphic meter in the circuit of the motor driving the exhaust fan and he began the test with the ducts at all of the machines open. Then he closed the ducts on each machine in succession. With all ducts open, the consumption was 9.6 kw. With all ducts closed, it was 5.1 kw. With the ducts open on only the machines in use at time of the test, the load was 6.3 kw —a saving of 35 per cent, made by closing the ducts on the idle machines.

Frequently no provision is made for conveniently closing exhaust ducts—so the workers habitually leave them wide open. As indicated, power consumption can be reduced by changing this habit and requiring each suction inlet to be tightly closed when not in use.

Reducing Compressed Air Cost

From one of the large shipbuilding plants comes an interesting application of graphic instruments used to reduce the cost of supplying compressed air to the plant. Since the required volume of compressed air varies widely at different times, several compressors of different capacities are installed and an effort is made to keep the capacity in operation roughly proportional to the requirements.

The compressors are equipped with unloaders, and the ratio of "loaded" to "unloaded" time, as shown by a meter chart, is a good measure of the ratio of operating capacity to air requirements. Graphic wattmeters in the circuits of the motors driving the compressors give records which show the loaded and unloaded periods. For instance, if a 2500 cu ft compressor operates unloaded 60 per cent and loaded 40 per cent of the time, it is obvious that the load is about 1000 cu ft and that a 1200 cu ft machine may be expected to carry the load with a safe margin.

A graphic chart showed a mixed load, including a 2500 cu ft, 450 hp compressor operating at nearly full load from 12:45 to 1:00 p.m. The chart showed that the air requirements gradually diminished, as indicated by the increase in the unloaded time. Finally, at 3:30 p.m., the load had diminished to the point where the large compressor could be shut down and a smaller one put on the line to carry the load. Since the efficiency of both motor and compressor is the highest near full load, power is conserved by adapting compressor capacity to the load requirements.

[Concluded on p. 383]

*President, Esterline-Angus Co.

Fig. 2—This is a graphic chart of a pump in distress. Often breakdown can be prevented by discovering such conditions and remedying them before failure occurs

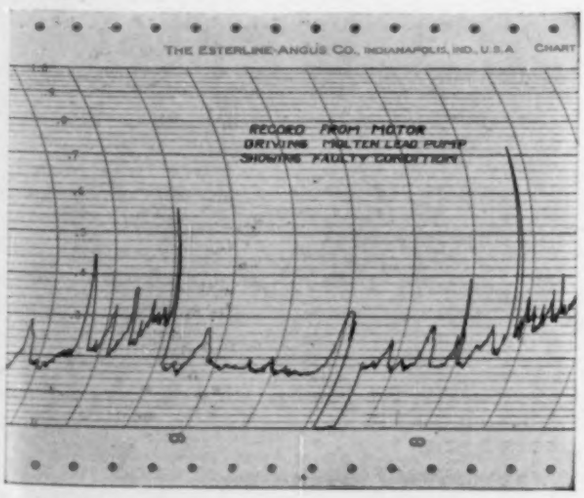
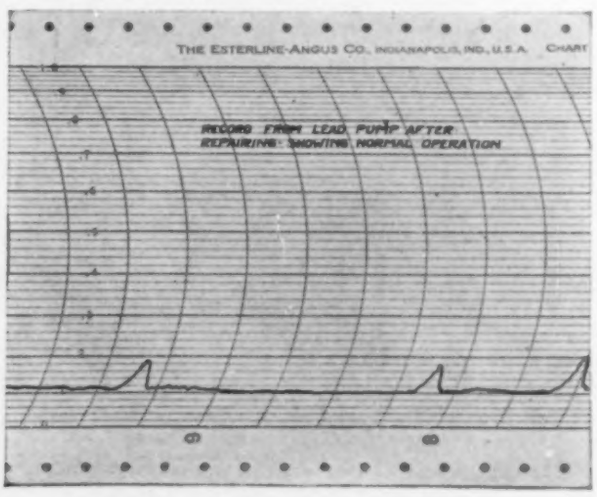

THE ESTERLINE-ANGUS CO., INDIANAPOLIS, IND., U.S.A. CHART
RECORD FROM MOTOR
DRIVING MOLTEN LEAD PUMP
SHOWING FAULTY CONDITION

Fig. 3—After the pump has been repaired and put back into service, this chart was made. It shows that everything is working smoothly again. Graphic tests of equipment will determine when machines need attention



Piping and Heating Maintenance Costs

are included in these forms developed by Plant Engineer Alfred Vaksdal for summarizing expense of plant steam

BECAUSE he believes that the advantages of good plant maintenance are obvious but the importance of its cost has not been fully appreciated, Alfred Vaksdal, plant engineer at the Corning Glass Works, has worked up record forms which may be used to set up a control for maintenance items, and to furnish an incentive for improvement. As automatic operation of plant equipment increases, direct labor is reduced—but maintenance increases. Therefore, good maintenance records become a necessity.

For some plants the forms (reproduced here) may not furnish enough detailed information. For others, too many items may be included. They have therefore been set up on a decimal classification basis so that items may be added or taken out as may be necessary to fit the individual case.

The Report of Steam Generation furnishes a form for summarizing the major items of steam and power costs. The Maintenance Detail Sheet is a break-down of Item 9 of the steam generation report and includes classifications for plant piping and heating services.

REPORT OF STEAM GENERATION

Plant_____ Engineer_____ Period_____

No.	ITEM	UNIT	UNIT COST	NUMBER OF UNITS	TOTAL COST (EVEN DOLLARS)
1	Electricity	Kwhr	$		$
2	L.P. Steam	M Lb			
3	Coal	Tons			
4	Oil	M Lb			
5	Gas	M Cu Ft			
6	Water	M Lb			
7	Other Materials				
8	Operating Labor				
9	Maintenance				
10	Supplies & Misc				
11	Supervision				
12	Total Operating Cost				
13	M Lb Net Steam Delivered				
14	Unit Cost per M				

OPERATING DATA

15 Steam Press_____. 16 Superheat_____. 17 Briefly describe Fuels (include Btu)_____

18 Describe boiler units very briefly. Give heating surface, how fired and whether economizers or air heaters are used. Omit fine details

19 Net boiler plant evaporation per pound of coal as fired _____

20 Same from and at 212 F°_____. 21 Net boiler plant efficiency _____

9.0 MAINTENANCE DETAIL SHEET

TOTAL COST $_____

Total cap. of plant _____ bhp

Max. rating _____ % Operating Time _____ hr

Av. rating _____ % Load Factor _____ %

Total steam delivered _____ lb

Maintenance per M lb delivered _____ ¢

Remarks _____

			ITEM	No.	CAPACITY	COST L	COST M	SUB TOTAL	TOTAL	UNIT COST
9.1			STEAM GENERATORS							
	.01		Boilers							
	.02		Soot Blowers							
	.03		Superheater							
	.04		Economizer							
	.05		Valves							
	.06		Meters & Gages							
	.07		Blow Down							
	.08		Miscellaneous							
9.2			Boiler Feed System							
	.01		Pumps							
	.02		Heaters							
	.03		Regulators							
	.04		Piping							
	.05		Meters & Gages							
	.06		Valves							
9.3			COMBUSTION SYSTEM							
	.01		Furnace							
	.02		Fuel System, Coal							
		.001	Handling Machinery							
		.002	Stokers							
		.003	Grates							
	.03		Fuel System, Gas, Oil or powdered fuel							
		.001								
		.002								
		.003								
	.04		Draft System							
		.001	Fans & Ducts							
		.002	Air Preheater							
		.003	Breeching & Stack							
	.05		Ash Handling System							
9.4			SOFTENING PLANT							
	.01		Tanks & Softening Bed							
	.02		Piping							
	.03		Pumps							
	.04		Filters							
9.5			STEAM SYSTEMS							
	.01		High Pressure							
		.001	Piping & Covering							
		.002	Regulators & Meters							
		.003	Valves & Traps							
	.02		Low Pressure							
		.001	Piping & Covering							
		.002	Regulators & Meters							
		.003	Valves & Traps							
	.03		Heating System							
		.001	Piping & Covering							
		.002	Valves & Traps							
		.003	Radiators & Unit Heaters							
		.004	Vacuum Pumps & Return Lines							

Chicago Board of Trade members and employees came through the winter and spring with a health record of practically 100 per cent freedom from pneumonia, influenza and common colds. The explanation lies in the fact that the trading hall and the executive offices are air conditioned, says Peter J. Niemann, exchange hall supervisor.

View of one of the aisles showing branch feeder and return line at left and piping for individual molds at right

High Pressure Hot Water System Heats Molds at Plastics Plant

BY installing a high pressure hot water system for heating molds in its reconstructed plastics plant at Pittsfield, Mass., the General Electric Co. has realized a saving in fuel of something like 39.5 per cent over the older method of heating the molds by steam. This installation has been in operation since last August, so there has been plenty of time to confirm this figure. The job is a complete one; it involved the installation of new oil fired boilers and entirely new piping for the 324 hydraulic presses, ranging in size from 2½ to 36 in. ram diameters, or total pressures on the rams of from 5 to 1000 tons respectively. The set-up is for an ultimate of 10,000,000 Btu per hr, the present consumption of heat being about half that amount.

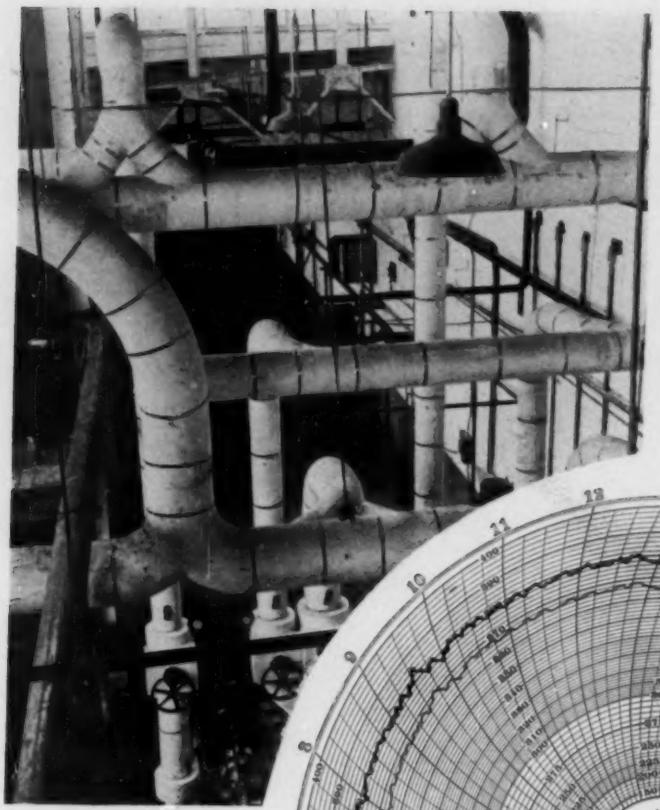
An analysis of the per cent savings from using the high pressure hot water for heating the molds has been given by the construction engineer for the plastics department as follows:

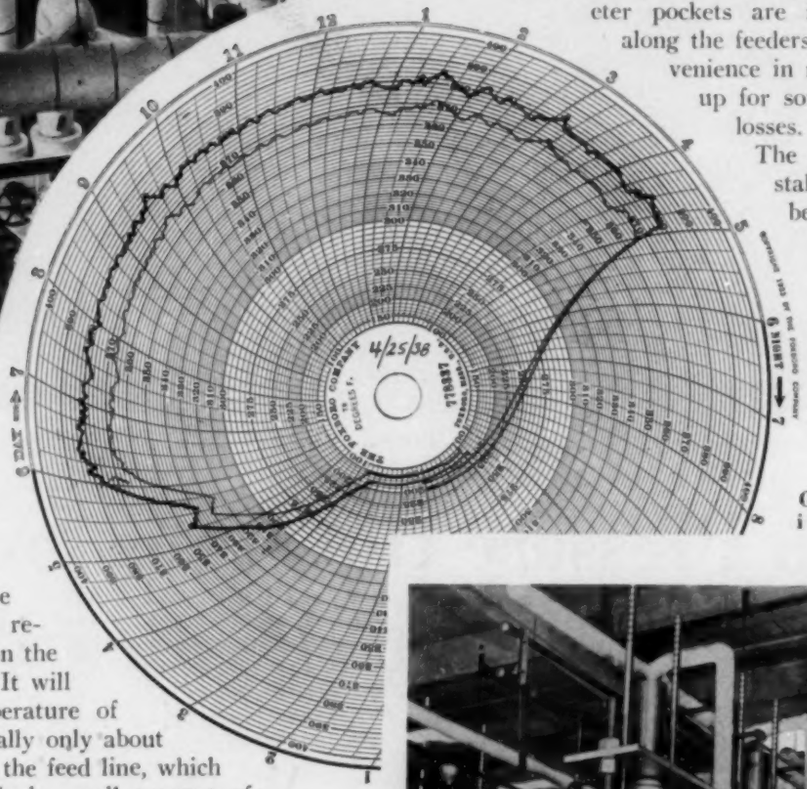
1. No loss due to flashing of high pressure condensate to low pressure steam	17.5%
2. Closed system without leaks or bleeding	3.0%
3. No traps or orifices blowing through, no bypasses around the traps	12.0%
4. Complete insulation of both feed and return lines	5.0%
5. No overflow or flashing of water to steam in hot well or feedwater heater	2.0%
Total	39.5%

In addition there are the further advantages of less make-up water or none, resulting in economies from reduced water bill, cost of heating feedwater, less scale in boilers, less frequent blowdowns, increased boiler efficiency, and less boiler cleaning.

By **F. A. Westbrook**

A high pressure hot water system (instead of steam) furnishes heat to the molds of the hydraulic presses at G. E.'s plastics plant at Pittsfield, Mass. The saving—confirmed by operation since last August—is said to amount to nearly 40 per cent. . . . Features of the system and the piping layout are described here. . . . The method is applicable in many industries having process heat requirements





Above—Headers above circulating pumps. *Right* — Temperature chart for feed and return lines, illustrating evenness of the heating

The evenness of the heat is shown by the chart taken from the recording thermometers on the feed and return lines. It will be seen that the temperature of the return line is generally only about 11 deg less than that of the feed line, which means that a comparatively small amount of fuel needs to be burned to keep the temperature of the feed line up to the required average of about 380 F. The advantage of the system lies in its ability to recover fully the unused heat. Also, for a given pressure and volume, water has much more heat stored in it than steam.

The Piping

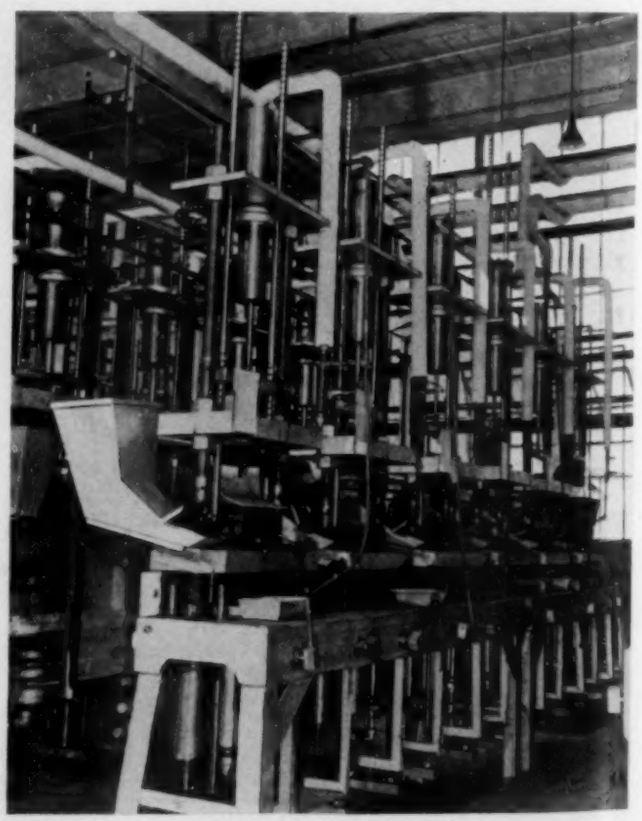
The new piping is all of seamless steel tubing. All joints are welded as are all valve flanges, to which the valves are bolted. Air bottles have been provided at about 12 high points on the lines to reduce trapped air to a minimum.

The piping is suspended so as to keep it at least 6 in. from any other object in case of movement. No such movement as this has ever taken place, doubtless because of careful operation, but the precautionary measure was taken nevertheless. In some cases the lines are suspended from brackets and in others from the roof trusses, by means of rods and clamps around the pipes. Standard 85 per cent magnesia pipe insulation is used, 2 in. thick for the mains and 1 in. thick on the branches.

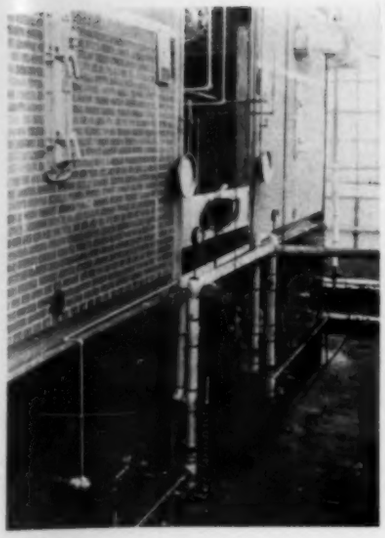
Except for the small 1¼ in. take-offs at the presses, there are no bull headed T connections. There is always a Y connection, which is a pipe on the larger sizes to help with the flow, and a blank for the smaller sizes. The reason for this method of construction is of course for the primary purpose of providing strength and therefore permanence, which naturally also means low maintenance cost and continuity of service.

The 8½ in. mains for the feed and return lines have expansion bends about halfway between the boilers and the anchorage where four branches are taken off for the service of the molds. Each of these branches also has two expansion bends, or loops. Thermometer pockets are installed at intervals along the feeders and returns for convenience in making tests to check up for sources of possible heat losses.

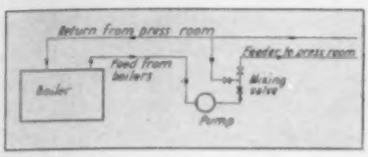
The description of the installation would hardly be complete without at least a brief reference to the boilers and connections at the circulating pumps. One of the advantages of the



Group of six small molds individually heated



Two oil fired boilers heat the water



Piping connections at boilers and pumps

system is that any standard type of industrial boiler may be used. The two used for this job are fired with oil and operate at 200 lb pressure. The hot water is drawn directly from the drums of these boilers and circulated through the system; after heating the molds, it is returned to them, the whole system being fully enclosed. The water level is kept higher in the boilers than is usual for steam generation as there is no need of leaving space for a surplus accumulation of steam in case of sudden withdrawals, and it is desirable to have as large a volume of hot water as possible to act as a heat stabilizer.

The piping connections at the boilers are shown diagrammatically in a sketch. The mixing valve is controlled manually, as is the boiler combustion control at the present time. There are two circulating pumps, each driven by a 40 hp motor, one of them being for standby service. They are of the rotary type, have a capacity of 500,000 lb per hr each, and raise the pressure to 230 lb per sq in., which is enough above boiler pressure to insure proper circulation.

This installation is the first of its kind in an American plastics molding plant, although the system of so-called "superheated" water is fairly widespread in Europe. The scheme is applicable to many industries requiring process heat, such as the ceramic industry for making tile and brick, the food industry for dairies, starch plants, in the textile industry for dye houses, finishing works, etc., for the paper industry, in the chemical industry for the making of soaps, drugs and so forth —to mention only a few.

Graphic Records—

[Concluded from p. 379]

Keeping Compressors Out of Step

A large dairy takes particular care to eliminate useless power consumption and to keep the maximum demand in each of its plants within prescribed limits. Each plant is equipped with a graphic wattmeter which makes a record of the power load.

In one of these plants are two ammonia compressors, each driven by a synchronous motor. A part of the time both machines are operated; at other times but one is used. Occasionally the chief engineer would receive from the operating department a chart indicating a widely swinging load.

This unusual condition was analyzed and the cause found to be that the operator had synchronized both compressor motors so that the machines were on the compression stroke at the same instant. Since both motors operated at the same speed, the peak loads were combined, causing the wide swings in the power load. Engineers are now instructed to synchronize the motors with the compressors out of step, so as to stagger the peaks.

Needless Expense of Motor Starts Reduced

A company making roofing and paper products has a 75 hp motor driving a water supply pump. When a survey of plant equipment was made with graphic instruments, it showed frequent starting of some of the equipment.

The water level controls were found to be set to maintain a head within a 2 ft variation, resulting in frequent starting of the pump, excessive wear and tear on the control equipment and unnecessary peak loads. The controls were changed so as to permit a head variation of 8 ft, and the chart obtained after the change was made showed the demand was reduced. Much less maintenance on the starting equipment was required thereafter.

Other Examples

Many more examples, involving equipment of almost every type, could be cited were space available. For example:

(1) A city engineer suspected a large centrifugal pump unit driven by a 1650 hp motor was not as efficient as it should be. A graphic wattmeter in the motor circuit gave data on the relation of power consumed to water pumped, led to the discovery of a 6 in. drainage pipe that was open and had been for some time. The water escaping through it accounted for the low efficiency.

(2) Charts of the night electrical load in a plant caused discovery of a 3 hp motor in the boiler room driving a vacuum pump that was drawing approximately 5.4 kw, which seemed excessive. Examination showed a permanent ground in the windings and after repair the consumption was only 1.5 kw. It was estimated the motor had wasted over $4000 worth of electricity in the seven years it had been operating.

(3) An exhaust fan motor in a paper mill would trip out intermittently, and to reduce the number of shutdowns someone would block the circuit breaker and a burned out motor would result. A portable graphic wattmeter in the motor circuit aided the engineer to locate the difficulty—paper would become wound up in the blades of the fan and the motor had to operate at an overload until the fan cut itself free.

(4) The engineer of an ice manufacturing plant increased production 13 per cent by putting in a graphic meter which records the "harvesting" schedule and the accuracy with which the schedule is followed.

Control of Air-Borne Organisms

Various Industrial Applications of Filters Reviewed

PRIOR to 1922, most of the better grades of gelatin for photographic work were imported from Holland. Our domestic gelatins became "cloudy" after a short time. It was understood, of course, that this was due to the presence of bacteria in the air used for drying gelatins but the problem of producing air which was adequately free from such impurities in commercial quantities was a new one. In 1922, the author was connected with the installation of air filters of the viscous impingement type in a large midwestern gelatin plant under a guarantee to produce air which was 99 per cent free from air-borne organisms. This installation was successful and the manufacturer was enabled to produce gelatin which was equal to or better than the imported product.

By H. C. Murphy*

Mr. Murphy reviewed several recent studies on bacteria in air in an earlier article, particularly the air pollution investigations in New York City which gave figures on the number of bacteria in air in schools, subways, non-ventilated and ventilated theaters, streets, and parks. A bacteriological survey of over 200 telephone booths was also summarized. . . . This month, various applications of filtration to control air-borne organisms in different industries are cited

Gelatin and similar products are priced largely on the number of bacteria in each sample, there being a difference of as much as 35c per lb between different grades. There are, of course, other factors in the pricing of gelatin but the bacteria count is a most important item. Having successfully handled this problem, other plants manufacturing photographic dry plates, gelatin capsules, powdered milk, sliced bacon, and the like, were equipped to the considerable improvement of the products.

Producers of powdered milk found that with manufacturing processes controlled by bacteria filters they were able to produce a product which would withstand severe climatic conditions, thus greatly broadening their field.

Meat packers had a very troublesome problem in the returns from "moldy" bacon. The utmost cleanliness was insisted upon in their bacon slicing departments. The tables and other equipment were kept absolutely clean. The girls were dressed in white but despite every precaution the moldy bacon persisted. After the installation of bacteria air filters the mold problem was eliminated.

A large producer of nut butter installed bacteria filters of the viscous impingement type and found that he reduced the mold spore count 99.6 per cent, thus making it possible to eliminate rancid butter. The filters were installed not only in the plenum chamber but on the duct outlets in the drying room. This prevented the entrance of spores into the ducts when the fan was shut down and eliminated a possible source of contamination.

A manufacturer of surgical sutures installed a double filtration system using a semi-viscous bacteria filter and reported as follows: "Vacuum cleaner dust was repeatedly sifted through a fine mesh screen until only the very finest particles remained. This dust was found to produce approximately 2,000,000 colonies on culture plates per gram of material by standard test methods. One gram of this dust was divided into eight equal parts, each of which was blown on the filters at 1 hr intervals while the apparatus was running. Gelatin plates exposed for 30 min periods during this eight hours showed but 12 colonies in all as against 2,000,000 potential colonies placed on the filters in the form of grit, certainly much harder to remove from the air than the usual linty type of dust encountered. The rate at which this dust was placed on the filter was estimated to be approximately 1000 times the rate at which the filter would be called upon to handle a similar quantity in our laboratory.

"A second test, using two of the most resistant types of bacteria, (Bacillus Subtilis and Staphylococcus Aureus) was then made and on 68 plates exposed periodically over a period of 933 hr, only 14 colonies developed."

The director of the biological laboratories of a large midwestern manufacturer of biological and pharmaceutical products outlined his investigations as to the efficacy of bacteria filters in removing air-borne organisms as follows: "Three-inch petri dishes containing layers of agar agar culture medium were exposed for 30 min periods at 12 stations about the room. After 30 min exposure the plates were covered with a sterile cover and incubated at 37 C for 24 hr. A count of the colonies on each plate, after this incubation period, gave an average of 1.75 colonies to a plate." His report stated that they considered this to indicate a very good condition of the air as far as bacteriological content was concerned, when it is remembered that during these 30 min periods there were four people working in the laboratory and the doors on each side of the room were opened and closed a number of times during the interval.

Effectiveness of Bacteria Filters in Breweries

In April, 1937, the Wahl-Henius laboratories, who specialize in brewery problems, reported to a Seattle brewery that of 10 air samples taken in an installation of semi-viscous filters using bacteria media, the maximum bacteria count for 72 hr incubation was three colonies. The report stated that for all practical pur-

*Vice-President, American Air Filter Co., Inc. Member of Board of Consulting and Contributing Editors. Mr. Murphy is chairman of the committee on air cleaning of the American Society of Heating and Ventilating Engineers.

poses the air tested might be considered free from air-borne bacteria.

Today practically all modern breweries utilize bacteria filters and a great deal of valuable data have been developed in this field. In a paper read before the annual convention of the master brewers' association in 1936, H. R. Henius[1] described investigations of the relative efficiencies of the viscous filter and clean, absorbent cotton. He stated: "The figure (not reproduced here) shows how effective and necessary this viscous coated type of filter really is. Both of these petri dishes were exposed on the same day for the same length of time,

Viscous unit filter installed in a brewery for control of air bacteria

namely 5 min, at a distance of about 2 ft from the filtering unit. The petri dish at the left represents a test made with both viscous coated filters and cotton in use and you will please note that there are no colonies whatsoever on this dish. The dish at the right represents an identical test made, however, with the viscous coated filter removed; and here you will note innumerable colonies, and I might add that these colonies represent an unusually virile type of cocci bacteria which thrives in wort.

"From one of our scientific stations I learned that the best medium for making the petri dish test for this kind of work is nutrient agar, which has a pH of 6.6. Before I had the viscous coated filter, I used a wort agar for testing the quality of the air, with a pH of about 5.1, and no colonies developed. When some difficulty was encountered I learned that the cause of my troubles was the quality of the air, and this was proved by making a test with nutrient agar. It was then found that the same air which appeared perfect with wort agar was very definitely imperfect as proved by the innumerable colonies on the petri dish with nutrient agar. The viscous coated filter was then installed as a final filtering unit, and since its installation absolute purity of yeast has prevailed.

"Incidentally, the petri dish showing the colonies was only incubated 24 hr because with the usual 48 hr incubation the colonies would have covered the entire surface and, therefore, I could not have shown the difference to the same advantage. . . . It has been suggested that this type of filter will bring oil into the wort. After two years' experience I can assure you that this does not happen."

It is my opinion that if specially treated bacteria filter media were substituted for the material used at present, double filtration would be unnecessary. Untreated filtering fabrics, however, often create ideal locations for the propagation and dispersion of bacteria.

In the September, 1927, *Brewers Technical Review*, E. H. Froese described the installation of air filters for the control of air-borne organisms in a Denver brewery: "The filters are of the automatic viscous type. A special oil designed for this particular application is used. Since this tempered and filtered air is also carried to the room which houses the wort coolers, carrying this room under constant pressure when a brew is being cooled over the coolers, it has never been found necessary to cease operations even during severe dust storms. These continuous oil filters positively remove all dust, dirt and bacteria."

Use in the Food Industry

In 1924, Dr. E. V. Hill conducted tests at the Union Stock Yards, Chicago, to determine the efficiency of viscous impingement filters. His report stated: "Tests to determine the efficiency of a filter in the removal of bacteria from the air by exposing standard agar plates are not entirely satisfactory, owing to the fact that it is very difficult to prevent contamination of plates exposed on the clean side of the filter in strong currents of air. Dust particles with their attendant bacteria are necessarily blown from the hand of the experimenter, plate cover, etc., consequently efficiencies shown by this method are usually lower than would be anticipated. The efficiency of the filter in bacteria removal as shown by this test method was 98.01 per cent."

Thirty years ago mushroom growing was almost a matter of pure chance. In 1904 an investigator in West Chester, Pa., found that mushrooms—a fungi developed from air-borne spores—could be successfully cultivated by a process which almost approaches laboratory technic. Both viscous and semi-viscous bacteria filters are used in connection with the process to prevent contamination by unwanted air-borne organisms.

In 1928 bacteria filters were installed in a tomato ripening plant at Detroit. Fruits such as bananas, tomatoes, etc., picked green and artificially ripened, are subject to considerable spoilage due to air-borne organisms in the air which is forced over the ripening fruit. Oxygen is absorbed during the ripening process and ethylene gas, chlorine gas, CO_2 and various other gases are used instead of or in connection with the air to hasten the ripening. It is of the utmost importance, however, that

[1]Henius, H. R., *Brewers Journal*, Oct. 15, 1936.

the air be free of foreign organisms if extensive spoilage is to be avoided. This process is in rather general use at the present time.

It is interesting to note how many of our new "discoveries" bear out the hard-headed but unexplainable knowledge of our ancestors. From time immemorial we have heard that "one bad apple will spoil a barrel." Warehousemen have observed for years that there is a sort of communal life or herd quality in apples when stored together. They tend to and indeed do ripen or rot at much the same rate.

In an address made before the British Association of Refrigeration, Sir William Bates Hardy stated "A stream of air which is passed over an apple contains some subtle emanations which profoundly influence other vegetable forms. Potatoes placed in the stream either do not sprout or if they do the sprouts are mis-shapen dwarfs, more like warts than anything else. Bananas are excited to a much more rapid ripening than ordinarily. It is apparently only the ripe or over-ripe fruit which set up these changes, although the effect on unripe apples is very noticeable as they are forced in this way to mature much more rapidly." It is possible that this is another manifestation of the effect of air-borne organisms.

Development of Air Cleaning Devices

The development of mechanical control for air-borne organisms is closely associated with the development of air cleaning devices. The successive steps in the art of air cleaning have been entirely logical. First came mechanical cleaning or removal of the air-borne solids with cloth or metal screens. This served for certain purposes but it was found to be far from an adequate solution of the problem. It stopped the large dirt but the smaller particles went through. For instance, ordinary commercial cement is guaranteed by the manufacturer to have passed through 100 mesh screen. It is apparent that the 40 mesh screen ordinarily used in general ventilation work allows a large part of the dirt to pass. On the other hand, by its very nature, a screen fine enough to remove small dirt particles soon clogs and obstructs the air flow.

Later "coke washers" were tried in commercial work. They were open to certain inherent objections and were never commercially accepted. In an article written by Samuel R. Lewis and Dr. E. V. Hill on air washer experiences it was stated: "Wire baskets or screens filled with coke and subjected to a spray of water were tried and while they arrested much of the dirt, the dirt stayed in the coke and wonderful crops of bacteria would grow in the favorable conditions created and appalling odors would emanate from the foul coke beds."[2]

Later came the development of the air washer, which overcame many of the objections to previous methods of air cleaning. It served effectually as an air cleaner and in fact still serves in cases where soot, smoke, carbon and similar particles insoluble in water are not a considerable factor. Attempts have been made to utilize commercial air washers in the control of air-borne organisms but this has not been altogether successful as the effectiveness of an air cleaner in this respect is closely associated with its efficiency in removing other impurities from an air stream. Commercial types of air washers are recognized as being relatively inefficient against certain forms of dust.

The A.S.H.V.E. *Guide* states[3] "for non-microscopic wettable dust air washer efficiencies average about 50 per cent unless the concentration of dust is high. Its effectiveness in removing greasy microscopic dust is practically negligible."

It was also found that under certain operating conditions air washers were a breeding place for bacteria and actually caused bacterial contamination of the air instead of purifying it.

Prof. G. L. Larson presented a paper at the annual meeting of the American Society of Heating and Ventilating Engineers in 1916. He stated[4]: "Bacteria Tests: These tests showed some startling and unexpected results. When using recirculated water the washers supplied bacteria to the air instead of removing them and even when using new water continuously did not show any marked efficiency as a bacteria remover. . . . The sand filters showed conclusively that the washer delivered bacteria to the air. The four sand filter tests checked very closely and showed an increase of bacteria in a ratio of about two to one. . . . At the same time that the sand filter tests were made, a series of petri plates were exposed for the same length of time on each side of the washer. These also show conclusively that the number of bacteria in the air is increased by the washer, but at a very much higher ratio than shown by the sand filters."

More recent investigations on this subject published by Carswell, Doubly and Nason[5] seem to corroborate the studies of Professor Larson and others very closely but indicate the possibility of more efficient results by the introduction of germicides in the water used in the air washer.

W. F. Wells[6] has shown that an air washer using polluted water distributed bacteria to the air and caused bacteria contamination of the air even in remote parts of the building.

Further progress in the design of air cleaning devices led to the development of the oil coated or viscous impingement type of air filter. The trapping of dust on viscous coated surfaces is not a new development. The viscous coated, hair lined surfaces of the human nostril are effective against certain forms of air impurities.[7, 8]

Various forms of the viscous impingement air filter have been developed but in general they consist of a series of extended surfaces coated with a viscous charging liquid having germicidal properties. The air-borne organisms trapped in the filter are prevented from further activity by the chemical action of the charging liquid and also by its mechanical action in forming a film about the organism which prevents further subdivision and activity.

The semi-viscous type of bacteria filter usually consists of cellulose sheets treated to insure germicidal action; these sheets, arranged in various forms to secure

[Concluded on p. 389]

[2]Lewis, S. R., and Hill, E. V., *Heating and Ventilating Magazine*, June, 1919.

[3]A.S.H.V.E. *Guide*, 1937, p. 218.
[4]Larson, G. L., A.S.H.V.E. *Transactions*, Vol 22, 1916.
[5]Carswell, T. S., Doubly, J. S., and Nason, H. K., *Industrial and Engineering Chemistry*, Jan. 1937.
[6]Wells, W. F., Mass. Dept. Health, p. 166, 1934.
[7]Lehmann, G., *Journal Ind. Hygiene*, 17, 37, 1935.
[8]Tourangan, F. J., and Drinker, P., *Journal Ind. Hygiene*, 1937.

Air Paths from Grilles

Throw of Air Stream from Straight Flow Grilles

By A. L. Greenlaw* and T. S. Hart†

ELIMINATION of drafts is a major air conditioning problem; aside from mechanical difficulties, drafts are probably the cause of 90 per cent of complaints.

The question of eliminating drafts is primarily one of acquiring a clear conception of the performance of all types of outlets so that they may be properly located, and the selection of the correct number, type, and size of outlets to satisfy each particular condition. To do this, however, workable laws must be established so that the distance an air stream will travel, and the amount it will rise or fall when there is a temperature differential, may be predicted and taken care of while the layout is being made.

It has been the purpose of the investigation described here to find definite laws for the throw and drop of an air stream and to determine what effect different types of grilles have on the air path.

Method of Making Tests

Fig. 1 diagrams the laboratory where these experiments were made. It was possible with this arrangement to test grilles up to 360 sq in. in area.

The quantity of air delivered to the test room was measured by the pressure drop across an orifice and was controlled by dampers and a bypass duct to the outside. The temperature of the air supplied was regulated by varying the amount bypassed by the cooling coils. Velocities were read with either an anemometer or a direct reading velocity meter, depending on whether an average or an instantaneous reading was desired. A potentiometer reading to ½ F and 21 thermocouples were used to measure the temperatures in the air stream. Since the back and side wall of the test room were movable, it was possible to vary the size and shape of the room according to requirements of the investigation.

In all the following tests, the point at which the velocity of the air stream did not exceed 100 fpm at any instant was taken as the end of the throw. The maximum velocity rather than an average was taken because the velocity near the end of the throw may vary as much as 100 fpm, due to pulsations apparently caused by a building up and releasing of pressure ahead of the incoming stream. Since the high points in these pulsations are what will be noticed by the occupant, the length of blow must be long enough to keep these surges from being objectionable.

The study on throw has been divided into two parts: (1) The effect of area, aspect ratio, and face velocity on the throw from ordinary straight flow grilles such as that shown in Fig. 2. (2) The effect of various types of grilles on the air stream.

Straight Flow Grilles

Three factors were thought of which might affect the throw of an air stream from straight flow grilles: (1) The size of the grille. (2) The velocity of the entering air stream. (3) The shape or aspect ratio (width / height) of the grille. To find the effect of each of these three factors, tests were made in which one of the factors was varied while the other two were held constant.

*Engineer, Tuttle & Bailey, Inc.
†Engineer, Tuttle & Bailey, Inc.

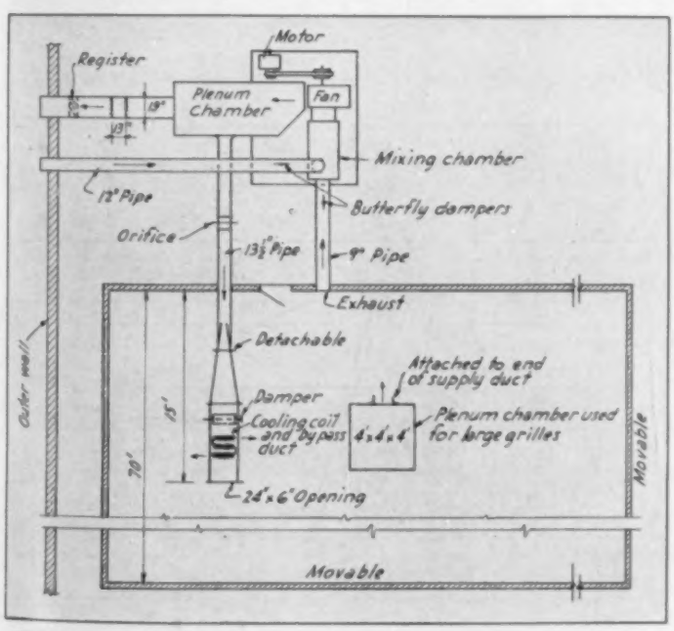


Fig. 1—Laboratory where experiments were made, with movable walls for varying size and shape of room

The authors have tested air conditioning grilles to determine from actual performance how the size, shape, and type of grille—and the velocity and temperature differential of the air leaving it—affect the air path. The results are reported here in part, and will be continued in a second article. . . . As a result of the studies, the following conclusions have been drawn: (1) The throw from a straight flow grille varies as the square root of the daylight area of the grille and as the face velocity. (2) The aspect ratio of a grille has no appreciable effect on the throw. (3) If the air streams from a grille are converged, it results only in cutting down the effective area of the grille. (4) Breaking the air stream up into jets has no effect on either the rate of mixing or the throw. (5) Fanning out the air stream shortens the throw, the amount depending on the amount of deflection. (6) The drop, for a given throw, of an air stream below room temperature varies about inversely as the face velocity and directly as the temperature differential. (7) Neither aspect ratio of the grille, breaking the air stream into jets, nor converging the streams have any effect on the drop of the air stream

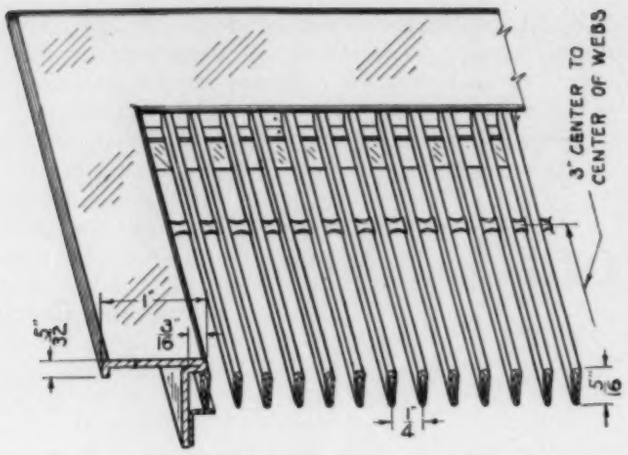


Fig. 2—Ordinary straight flow grille

Effect of Size on Throw—To test for the effect of size on throw, the face velocity and aspect ratio were held constant and the area varied. Table 1 gives results of these tests. When the square root of the daylight area was plotted against the throw, the result was a straight line, Fig. 3. In other words, if the face velocity and aspect ratio are held constant, the throw varies as the square root of the area.

Effect of Velocity on Throw—With a given grille the face velocity will vary as the number of cfm delivered through it. Since it was more accurate to determine the cfm delivered through the grille by measuring the pressure drop across the orifice than to determine the face velocity, the throw was plotted against the volume in cfm, Fig. 4. The result was a straight line, showing that the throw varies directly as the cfm delivered through the outlet. Since the velocity varies as the volume, it can, therefore, be said that for a grille of a given area and aspect ratio the throw will vary as the face velocity.

Effect of Aspect Ratio on Throw—The third factor that might affect the throw of a straight flow grille was its aspect ratio or shape. Considering only peripheral friction, one would expect the throw from two grilles of the same area but different aspect ratios to differ as their perimeters.

The throw from a 12" x 12" grille was checked against that from a 24" x 6" grille, with the same cfm being delivered through each. The results were as follows:

Grille size	12 x 12	24 x 6
Air volume	430 cfm	430 cfm
Velocity 4 in. in front of grille	650 fpm	650 fpm
Distance to point where anemometer was stationed	25 ft	25 ft
Anemometer readings	160 ft, 2 min	185 ft, 2 min
Static pressure behind grille	0.020	0.020

Believing the difference in perimeters to be the determining factor, it was expected that the throw from the 24" x 6" grille would be 4/5 as long as that from the 12" x 12", but as can be seen it is slightly more.

An air stream stops because it expends its energy by starting room air in motion. The stream from the 24" x 6" grille will undoubtedly lose more energy than the square grille due to peripheral friction; that is, it will pull more air along with it than the square one. The stream from the 12" x 12" grille, however, has a much blunter front than that from the 24" x 6" grille, and it will have a tendency to push the air ahead of it; whereas, with the longer grille, the air in front will tend to slip out by the edge of the stream. The thin air stream then, will have to pull more air along with it than the square one but will have to push less; for the square stream, the conditions will be reversed. These apparently about counterbalance one another so that the sum of the forces tending to stop the square stream are about equal to those tending to stop the longer one. In other words, for all practical purposes the effect of aspect ratio on the throw may be disregarded.

Length of Throw Formula—With straight flow grilles, we have found that there are two factors which affect the distance an air stream will travel; namely, the

Table 1—Effect of Size on Throw

Listed Grille Size, Inches	Actual Size of Daylight Opening, Inches	Square Root of Daylight Opening Area	Face Velocity, Fpm	Throw, Feet	Air Velocity at End of Throw, Fpm
30 x 6	29⅛ x 5⅛	12.5	900—950	55	15—100
20 x 4	19⅛ x 3⅛	8.0	900—950	35	15—105
15 x 3	14⅛ x 2⅛	5.75	900—950	25	15—100
10 x 2	10 x 2	4.47	900—950	20	25—100

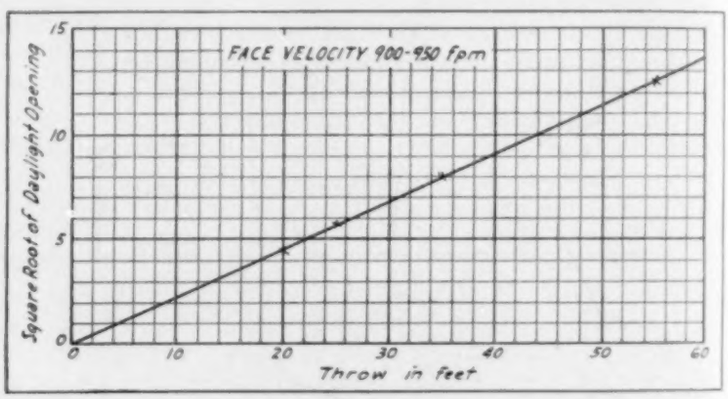


Fig. 3—With face velocity and aspect ratio constant, throw varies as the square root of the daylight area

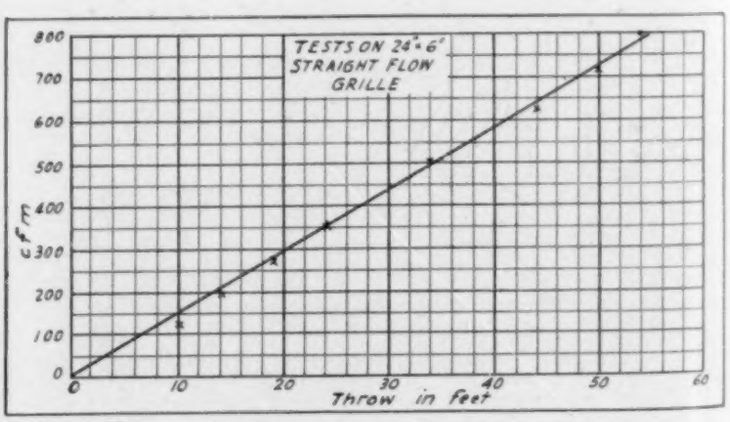


Fig. 4—Throw varies directly as the cfm delivered through the outlet

square root of the daylight opening area of the grille and the face velocity. Putting this in the form of an equation we have:

$$T = K_1 \sqrt{A}\, V \quad \ldots\ldots [1]$$

where

T = throw, ft;
A = daylight area of grille, sq in.;
V = face velocity, fpm;
K_1 = constant.

The following formula is well-known and is accurate enough for practical purposes:

$$\text{cfm} = K_2 A V \quad \ldots\ldots [2]$$

Solving for V and substituting in [1] the following formula is obtained:

$$T = K \frac{\text{cfm}}{\sqrt{A}} \quad \ldots\ldots [3]$$

The throw (T) can be found from the architect's drawing; the cfm is figured by the engineer himself; K can be found from experiments by the grille manufacturer. The only unknown in the formula is the area, which can be solved for.

To check the formula, the throws from grilles ranging from 6″ x 6″ to 36″ x 10″ in size and with several different face velocities were calculated, and the opposite wall of the test booth was moved to the point calculated as the end of the throw. The velocity in the stream a short distance from the wall and down it was checked with each grille, and the formula was found to hold true in every case studied.

[A second article will discuss face designs, factors controlling vertical path of air stream, and effect of different temperature differentials on air drop.]

Air Borne Organisms—

[Concluded from p. 386]

extended surfaces and low velocities through the media, are commercially available in a number of designs.

Air Pollution and Health

No attempt has been made in this paper to consider the relationship between abnormal air pollution and the spread of certain epidemic diseases.

Dr. J. S. Owens, in a paper read before the British Institution of Mechanical Engineers, has stated[9]:

"It has been gradually recognized of recent years that the impurities in the air which are injurious to health have no relationship to CO_2, but are, in the vast majority of cases, the suspended particulate matters. Of course, this does not mean that gas poisoning is impossible if a poisonous gas is emitted into the air, and there is no intention of dealing with this aspect of the question, but rather with what ventilation is required for ordinary inhabited houses and buildings. Diseases which are capable of being transmitted from one person to another are so transmitted by particulate matter. The most outstanding case of this kind is that of pulmonary tuberculosis, by the inhaling of dust containing tubercle bacilli or their spores.

"It is contended here that the whole outlook on ventilation has been unsound, and that instead of concentrating attention upon the amount of a harmless gas such as CO_2 in the air, it should be devoted to the concentration of particulate matter, and especially to pathogenic bacteria. Looked at from this point of view, the standard of ventilation should be one which keeps the concentration of dangerous suspended matter below a harmful point, or at least as low as can be conveniently arranged. It is true that this is a somewhat vague standard to fix, because the danger point for one person may be different from that for another, and indeed for the same person it may differ at different times, depending upon the capacity to resist infection."

Without attempting to evaluate the desirability of air cooling, humidifying or dehumidifying, it seems evident the elimination of dirt, dust and air-borne organisms secured with properly designed air conditioning can be of definite value to health. In the writer's opinion this phase of air conditioning has received too little attention.

[9]Owens, J. S., *Colliery Guardian*, Jan. 8, 1937.

Check Charts for Air Conditioning Calculations

By J. M. Dean*

A GENERAL introduction to this set of charts was given last month on page 321, and two forms—a comfort cooling survey analysis sheet and a guide sheet which serves as an index to the charts and provides a place to fill in the computed values—were published.

Chart 1 (on the next page) is self-explanatory.

Chart 2 is for determining U factors or heat transmission coefficients of walls built up of two or more kinds of materials. The combination of reciprocals in the usual fractional form is somewhat tedious and frequently the values are more or less guessed at. The vertical parallel nomogram at the right of this chart is for determining conductance values from tables of k values as found in the *Heating, Ventilating, Air Conditioning Guide*.

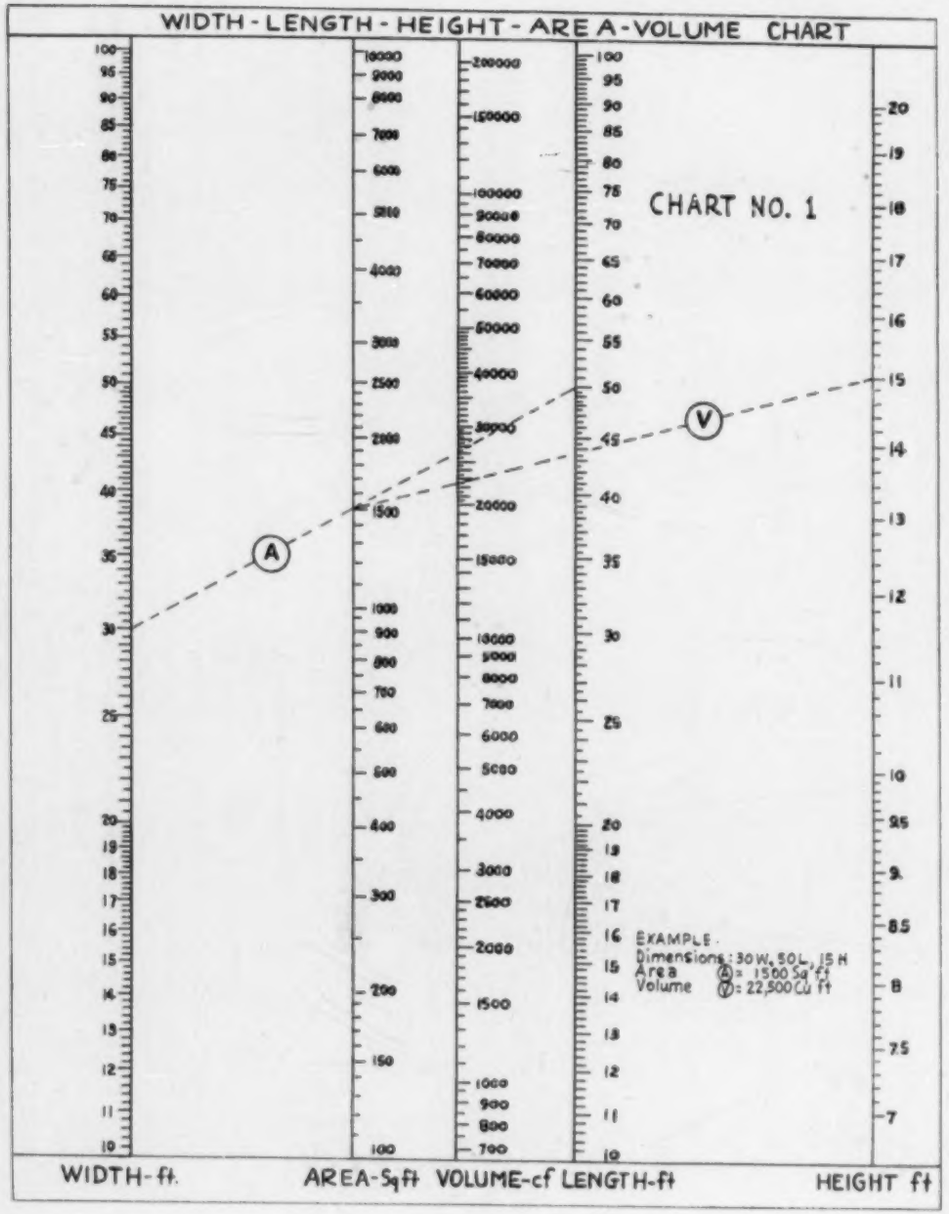
The V shaped nomogram at the left of this chart is for combining the conductance values, taken two at a time. The resulting U factor is indicated on the middle scale of whichever set of three converging scales is used, as indicated by the key on the diagram. It should be noted that each of the outer pair of the larger convergent scales is calibrated 0 to 100. When the outer scales are used at their face value, the middle scale should be read at its face value. However, the usual use of this nomogram will be to divide each of the calibrations on the outer scales by 10 in which case the middle scale readings should also be divided by 10. This is the only chart in the set in which proper decimal positions must be regarded. One or two trials in the use of this chart will quickly familiarize one with its operation.

The other charts and tables in this series will follow in order in future issues.

*Air Conditioning Engineer, Des Moines Electric Light Co. All rights reserved.

Check Charts for Air Conditioning Calculations

[See preceding page]



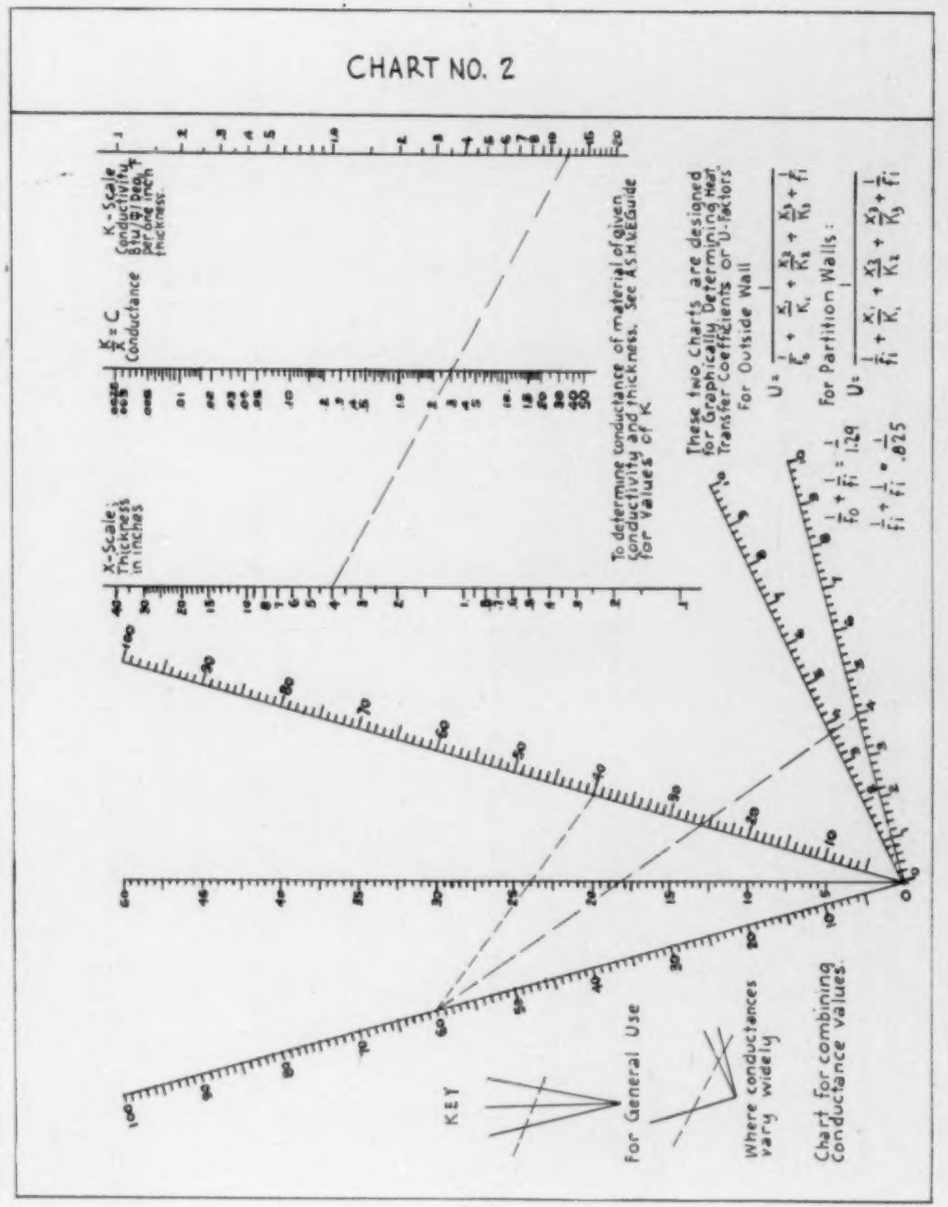


CHART NO. 2
K-Scale (Conductivity Btu/(ft²)(Deg. F) per one inch thickness)
K/X = C Conductance
X-Scale: Thickness in inches
To determine conductance of material of given conductivity and thickness. See A.S.H.&V.E. Guide for values of K
These two charts are designed for Graphically Determining Heat Transfer Coefficients or U-factors
For Outside Wall
For Partition Walls:
KEY
For General Use
Where conductances vary widely
Chart for combining Conductance values.

Figuring Solar Heat Gains of Buildings

By William Goodman*

Presentation of a new series of tables for estimating the solar heat gains of buildings in calculating air conditioning loads is continued this month. Use of the data for walls and roofs is explained

UNLIKE the tables for solar heat gains through glass that were discussed last month, the tables for computing the solar heat gains through walls and flat roofs list a fictitious temperature difference which, for convenience, is called the *solar temperature difference*. This solar difference is used in Equation 2 for computing the solar heat gain through a wall or flat roof.

$$H_s = AUD_s \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [2]$$

There can be no conduction heat gain through an opaque surface on which the sun is shining. The temperature of such a surface becomes higher than the temperature of the outdoor air. As a result, no heat can flow from the outdoor air to the surface; in fact there is always a flow of heat from the hot exposed surface to the outdoor air. However, if the surface were in the shade for a sufficient length of time, the ordinary conduction heat gain would take place. As will be shown later, it is convenient to compute only the amount by which the actual solar heat gain exceeds the ordinary conduction heat gain that would take place if the surface were in the shade. Hence, in preparing the solar temperature differences listed in Tables 1 to 8, the ordinary conduction heat gain was subtracted from the actual solar heat gain. For this reason, when these solar temperature differences are used in Equation 2, the answer obtained is not the actual solar heat gain through the wall or roof, but only the amount by which the total solar heat gain *exceeds* the ordinary conduction heat gain. In order to indicate that this quantity is less than the actual solar heat gain through walls and roofs, it is henceforth called the *excess* solar heat gain.

By using the excess solar heat gain instead of the actual solar heat gain, the nuisance is avoided of first having to separate the various areas of a building according to whether only solar or only conduction heat gains are to be figured. The conduction heat gain is first figured by means of Equation 1 for *all* of the exposed surfaces just as if they were all in the shade. Then the excess solar heat gains of only the surfaces exposed to the sun are figured by [2]. The total heat gain through the walls and roof is the sum of the conduction and excess solar heat gains. In this way, if necessary, the solar heat gain can be computed for several different times of the day without in any way altering or repeating the computations for the conduction heat gain alone.

The answer obtained for windows by means of the tables represents the *actual* solar heat gain through the windows; the answer obtained for walls represents the *excess* solar heat gain. To find the total heat gain through windows, the conduction heat gain must be added to the actual solar heat gain through the windows. In the case of walls on which the sun is shining, there is no conduction heat gain. However, in order to find the actual solar heat gain through a wall, the conduction heat gain must be added to the *excess* solar heat gain. Thus, for both windows and walls, the arithmetical procedure is the same—the conduction heat gain must be added to the value obtained by means of Table 1 to 8, though the reason for doing so is different for the two cases. For convenience, in the balance of this article the actual solar heat gain through windows is also sometimes called the excess solar heat gain though this appellation is not correct.

The time of day at which the solar heat gain through the walls and roofs is determined should be the same as the time of day when the solar heat gains through the windows are to be figured. In other words, all of the solar heat gains of a building are figured at the same hour of the day. The particular time of day should be selected when the *sum* of the solar heat gains through all of the windows, the walls, and the roof is a maximum, unless other considerations require the choice of another time.

The color and texture of a surface affect the amount of the impinging solar radiation that will be absorbed by the surface. A multiplying factor cannot be used for determining from the excess solar heat gain of a dark surface, the excess solar heat gain of a lighter colored surface. Hence, separate tables are given for dark and medium colored surfaces. Surfaces may be classified approximately as *dark* or *medium* as follows:

Dark surfaces: black, dark green, roofing materials.

Symbols

A = area of exposed surface, sq ft

D = difference between temperatures of outdoor and room air, degrees Fahrenheit ($D = t_o - t_R$)

D_s = solar temperature difference for walls and roofs, degrees Fahrenheit

H_c = heat gain due to conduction through exposed surface, Btu per hr

H_s = solar heat gain through glass or excess solar heat gain through wall or roof, Btu per hr

t_o = temperature of outdoor air, degrees Fahrenheit

t_R = temperature of room air, degrees Fahrenheit

U = over-all coefficient of heat transfer, Btu per hr, per sq ft, per degree Fahrenheit difference

*The Trane Co. Member of Board of Consulting and Contributing Editors.

Part 2. Part 1 was published in HEATING, PIPING AND AIR CONDITIONING, May, 1938, pp. 315-318.

Copyright, 1938, by William Goodman.

Medium-colored surfaces: red, brown, light green, gray.

Light colored surfaces seldom remain light long; they are soon darkened by dirt and weathering. For this reason, light colored surfaces are best figured as though they are medium colored.

The effect of sunlight shining through glass is usually instantaneous. There is no time lag, unless the sunlight falls on a heavy concrete floor which absorbs the heat and then returns it slowly to the room after the temperature of the floor has been raised. In the case of walls and roofs, before the heat of the sun can penetrate into the room on the other side, the temperature of the entire mass of the wall or roof must be raised. This usually requires some time. As a result, the solar heat gain through a wall may be due to the solar heat which impinged on the outside of the wall a few hours earlier. This is known as the *time lag* of the wall, and is discussed later. For walls and roofs of ordinary construction the time lag is about two or three hours. Thus, if the time lag is assumed to be two hours, the solar heat that reaches a room at say, 3 p. m. is due to the radiant energy that impinged on the walls and roof of that room at 1 p. m. The tables for walls and roofs are based on an instantaneous transmission of solar heat which actually does not exist. However, allowance can be made by figuring the solar heat gain through a roof at, say,

Solar Heat Gains Through Windows and Skylights and Solar Temperature Differences for Walls and Roofs

Table 5—35 Degrees Latitude

WINDOWS AND SKYLIGHTS

Solar Heat Gain
Btu per Sq Ft of Glass

A. Single glass—Bare

Time ↓	Direction Window Faces NE	E	SE	S	Flat Skylight	Time
6 a.m.	48	52	23		5	6 p.m.
7 a.m.	118	147	81		53	5 p.m.
8 a.m.	130	187	126		129	4 p.m.
9 a.m.	86	170	132	6	192	3 p.m.
10 a.m.	27	116	113	24	240	2 p.m.
11 a.m.		40	73	39	267	1 p.m.
12 M			23	46	277	12 M
	NW	W	SW	S	Flat Skylight	↑ ←

B. Double glass—Bare

Time ↓	Direction Window Faces NE	E	SE	S	Flat Skylight	Time
6 a.m.	42	46	17		2	6 p.m.
7 a.m.	102	129	66		37	5 p.m.
8 a.m.	107	163	103		107	4 p.m.
9 a.m.	61	144	107	1	166	3 p.m.
10 a.m.	11	89	86	9	210	2 p.m.
11 a.m.		20	47	19	235	1 p.m.
12 M			9	24	244	12 M
	NW	W	SW	S	Flat Skylight	↑ ←

WALLS AND ROOFS

Solar Temperature Differences
Degrees F

C. Dark Colored

Time ↓	Direction Wall Faces NE	E	SE	S	Roof	Time
6 a.m.	4	5				6 p.m.
7 a.m.	22	28	14		8	5 p.m.
8 a.m.	26	39	25		26	4 p.m.
9 a.m.	18	35	27		40	3 p.m.
10 a.m.	6	24	24	5	51	2 p.m.
11 a.m.		9	16	9	57	1 p.m.
12 M			5	11	60	12 M
	NW	W	SW	S	Roof	↑ ←

D. Medium Colored

Time ↓	Direction Wall Faces NE	E	SE	S	Roof	Time
6 a.m.	2	2				6 p.m.
7 a.m.	15	20	9		5	5 p.m.
8 a.m.	19	28	18		18	4 p.m.
9 a.m.	12	26	20		29	3 p.m.
10 a.m.	3	17	17	2	38	2 p.m.
11 a.m.		6	11	5	43	1 p.m.
12 M			2	7	45	12 M
	NW	W	SW	S	Roof	↑ ←

Table 6—40 Degrees Latitude

WINDOWS AND SKYLIGHTS

Solar Heat Gain
Btu per Sq Ft of Glass

A. Single glass—Bare

Time ↓	Direction Window Faces NE	E	SE	S	Flat Skylight	Time
6 a.m.	47	52	24		7	6 p.m.
7 a.m.	114	147	86		55	5 p.m.
8 a.m.	120	190	124	1	129	4 p.m.
9 a.m.	71	170	146	18	189	3 p.m.
10 a.m.	15	116	132	45	234	2 p.m.
11 a.m.		40	93	66	260	1 p.m.
12 M			40	74	269	12 M
1 p.m.			2			11 a.m.
	NW	W	SW	S	Flat Skylight	↑ ←

B. Double glass—Bare

Time ↓	Direction Window Faces NE	E	SE	S	Flat Skylight	Time
6 a.m.	41	46	18		3	6 p.m.
7 a.m.	98	129	71		39	5 p.m.
8 a.m.	98	166	113		106	4 p.m.
9 a.m.	47	144	121	6	163	3 p.m.
10 a.m.	4	89	105	24	205	2 p.m.
11 a.m.		20	66	41	228	1 p.m.
12 M			20	47	237	12 M
	NW	W	SW	S	Flat Skylight	↑ ←

WALLS AND ROOFS

Solar Temperature Differences
Degrees F

C. Dark Colored

Time ↓	Direction Wall Faces NE	E	SE	S	Roof	Time
6 a.m.	4	5				6 p.m.
7 a.m.	21	28	15		9	5 p.m.
8 a.m.	24	39	27		26	4 p.m.
9 a.m.	15	35	30	3	39	3 p.m.
10 a.m.	3	24	28	10	50	2 p.m.
11 a.m.		9	20	15	56	1 p.m.
12 M			9	16	58	12 M
	NW	W	SW	S	Roof	↑ ←

D. Medium Colored

Time ↓	Direction Wall Faces NE	E	SE	S	Roof	Time
6 a.m.	1	2				6 p.m.
7 a.m.	14	20	10		5	5 p.m.
8 a.m.	17	29	20		18	4 p.m.
9 a.m.	10	26	22	1	29	3 p.m.
10 a.m.		17	20	6	37	2 p.m.
11 a.m.		6	14	10	42	1 p.m.
12 M			6	11	44	12 M
	NW	W	SW	S	Roof	↑ ←

3 p. m. by means of the values in the 1 p. m. row of the tables.

Time lag depends on the material and mass of the wall; it may vary from zero for glass to 10 hours or more for a massive or heavily insulated wall. For walls and roofs of ordinary construction, the time lag is probably about from two to four hours. The time lag of insulated walls and roofs may be anything from about two hours up, depending upon the construction and the amount of insulation. Sufficient data are not available about typical types of construction to enable more definite values to be given.

Example 3: The building of Example 1 has walls of medium color of the following net areas (gross area minus glass area) facing in the directions noted: East wall: 725 sq ft. South wall: 1450 sq ft. West wall: 950 sq ft.

The roof is insulated, dark in color, and has an area of 13,000 sq ft. The over-all coefficient of the wall is 0.3, and of the roof, 0.15. Find the excess solar heat gain through the walls and roof. Assume that the time lag of the walls is two hours, and of the roof four hours. The north wall is a partition on the other side of which there is also a conditioned room.

Solution: The over-all coefficient of the walls is 0.3 and their time lag is assumed to be 2 hr. As the solar heat gain is to be computed at 3 p.m., the solar temperature differences should be read from the row for 1 p.m. in Table 6-D. The roof is insulated, dark in color, and has an area of 13,000 sq ft. Its

Solar Heat Gains Through Windows and Skylights and Solar Temperature Differences for Walls and Roofs

Table 7—45 Degrees Latitude

WINDOWS AND SKYLIGHTS

Solar Heat Gain
Btu per Sq Ft of Glass

A. Single glass—Bare

Time → / ↓	Direction Window Faces NE	E	SE	S	Flat Skylight	Time
6 a.m.	46	52	25		8	6 p.m.
7 a.m.	109	147	91		56	5 p.m.
8 a.m.	110	190	145	4	127	4 p.m.
9 a.m.	57	170	160	33	184	3 p.m.
10 a.m.	6	116	149	69	227	2 p.m.
11 a.m.		40	112	93	250	1 p.m.
12 M			59	102	260	12 M
1 p.m.			8			11 a.m.
	NW	W	SW	S	Flat Skylight	↑ ←

B. Double glass—Bare

Time → / ↓	Direction Window Faces NE	E	SE	S	Flat Skylight	Time
6 a.m.	40	46	19		4	6 p.m.
7 a.m.	93	129	75		40	5 p.m.
8 a.m.	87	166	123	1	104	4 p.m.
9 a.m.	35	144	134	16	158	3 p.m.
10 a.m.	1	89	122	44	198	2 p.m.
11 a.m.		20	84	66	220	1 p.m.
12 M			35	74	228	12 M
1 p.m.			2			11 a.m.
	NW	W	SW	S	Flat Skylight	↑ ←

WALLS AND ROOFS

Solar Temperature Differences
Degrees F

C. Dark Colored

Time → / ↓	Direction Wall Faces NE	E	SE	S	Roof	Time
6 a.m.	4	5				6 p.m.
7 a.m.	20	28	16		9	5 p.m.
8 a.m.	22	39	29		25	4 p.m.
9 a.m.	12	35	33	7	38	3 p.m.
10 a.m.		24	31	15	48	2 p.m.
11 a.m.		9	24	20	54	1 p.m.
12 M			13	22	56	12 M
	NW	W	SW	S	Roof	↑ ←

D. Medium Colored

Time → / ↓	Direction Wall Faces NE	E	SE	S	Roof	Time
6 a.m.	1	2				6 p.m.
7 a.m.	14	20	11		5	5 p.m.
8 a.m.	15	29	21		18	4 p.m.
9 a.m.	8	26	24	4	28	3 p.m.
10 a.m.		17	23	10	36	2 p.m.
11 a.m.		6	17	14	40	1 p.m.
12 M			9	15	42	12 M
	NW	W	SW	S	Roof	↑ ←

Table 8—50 Degrees Latitude

WINDOWS AND SKYLIGHTS

Solar Heat Gain
Btu per Sq Ft of Glass

A. Single glass—Bare

Time → / ↓	Direction Window Faces NE	E	SE	S	Flat Skylight	Time
6 a.m.	45	52	26		9	6 p.m.
7 a.m.	105	147	96		57	5 p.m.
8 a.m.	100	190	154	11	124	4 p.m.
9 a.m.	45	170	173	51	177	3 p.m.
10 a.m.	1	116	165	92	217	2 p.m.
11 a.m.		40	131	120	239	1 p.m.
12 M			77	129	247	12 M
1 p.m.			18			11 a.m.
	NW	W	SW	S	Flat Skylight	↑ ←

B. Double glass—Bare

Time → / ↓	Direction Window Faces NE	E	SE	S	Flat Skylight	Time
6 a,m.	39	46	20		5	6 p.m.
7 a.m.	89	129	80		41	5 p.m.
8 a.m.	77	166	132	3	102	4 p.m.
9 a.m.	24	144	147	30	152	3 p.m.
10 a.m.		89	138	66	189	2 p.m.
11 a.m.		20	104	93	209	1 p.m.
12 M			51	102	217	12 M
1 p.m.			6			11 a.m.
	NW	W	SW	S	Flat Skylight	↑ ←

WALLS AND ROOFS

Solar Temperature Differences
Degrees F

C. Dark Colored

Time → / ↓	Direction Wall Faces NE	E	SE	S	Roof	Time
6 a.m.	4	5				6 p.m.
7 a.m.	19	28	17		9	5 p.m.
8 a.m.	20	39	31	1	25	4 p.m.
9 a.m.	10	35	36	11	37	3 p.m.
10 a.m.		24	35	20	46	2 p.m.
11 a.m.		9	28	25	51	1 p.m.
12 M			17	27	53	12 M
1 p.m.			4			11 a.m.
	NW	W	SW	S	Roof	↑ ←

D. Medium Colored

Time → / ↓	Direction Wall Faces NE	E	SE	S	Roof	Time
6 a.m.	1	2				6 p.m.
7 a.m.	13	20	11		5	5 p.m.
8 a.m.	14	29	23		18	4 p.m.
9 a.m.	6	26	26	7	27	3 p.m.
10 a.m.		17	25	14	34	2 p.m.
11 a.m.		6	20	18	38	1 p.m.
12 M			12	20	40	12 M
1 p.m.			1			11 a.m.
	NW	W	SW	S	Roof	↑ ←

over-all coefficient is 0.15 and its time lag is 4 hr. As the solar heat gain is to be computed at 3 p.m., the solar temperature difference should be read from Table 6-C at 11 a.m. The excess solar heat through each of the surfaces is figured by Equation 2.

South wall: $1450 \times 0.3 \times 10$	=	4,350 Btu per hr
West Wall: $950 \times 0.3 \times 6$	=	1,710
Roof: $13,000 \times 0.15 \times 56$	=	109,200
Total excess solar heat gain	=	115,260 Btu per hr

The total heat gain of a building is the sum of the solar heat gains through the windows; the excess solar heat gains through the walls and roof; the conduction heat gains through the windows, walls, and roof; and the heat gains from all other sources such as people and lights.

The variation in the total heat gain of a building throughout the day is interesting. The computation of the total heat gain and its variation throughout the day are illustrated in the following example:

Example 4: Compute the total heat gain at all hours of the day for the building whose various glass and wall areas are given in Examples 1 and 3. In addition, assume that all of the windows are equipped with venetian blinds, and that the heat gain due to people, lights, and other sources is constant throughout the day and amounts to 40,000 Btu per hr. The difference in temperature between the outdoor and indoor air is 15 deg.

Solution: First, the conduction heat gain through all of the windows, the walls, and the roof must be found. From Example 1, the total area of the windows is $175 + 350 + 250 = 775$ sq ft. The over-all coefficient for glass is 1.1.

For the windows,

$$H_c = AUD \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [1]$$
$$= 775 \times 1.1 \times 15$$
$$= 12,790 \text{ Btu per hr}$$

From Example 3, the total net wall area for all of the exposed wall surfaces is $725 + 1450 + 950 = 3125$ sq ft.

For the walls, $H_c = AUD = 3125 \times 0.3 \times 15 = 14,060$ Btu per hr.

For the roof, $H_c = AUD = 13,000 \times 0.15 \times 15 = 29,250$ Btu per hr.

(The conduction heat gain through the walls and roof is figured as if the entire building were in the shade. The fact that the sun is shining on some of these surfaces is disregarded for the present).

Total heat gain of the building without solar heat is:

	Glass	= 12,790 Btu per hr
	Walls	= 14,060
	Roof	= 29,250
Total conduction heat gain		= 56,100 Btu per hr
People, lights, etc.		= 40,000
Sum of all heat gains except solar		= 96,100 Btu per hr

The total heat gain for any one hour of the day is found by adding 96,100 Btu to the actual solar heat gain through the windows and the excess solar heat gain through the walls and roof. For example, at 3 p.m., the solar heat gains are:

From Example 2, solar heat gain through windows with venetian blinds	= 29,280 Btu per hr
From Example 3, excess solar heat gain through walls and roof	= 115,260
Total excess solar heat gains	= 144,540 Btu per hr
Sum of all other heat gains except solar	= 96,100
Total heat gain at 3 p.m.	= 240,640 Btu per hr

The total heat gain at all other hours of the day was computed in the same manner and the results tabulated in the table below.

Time	Excess Solar Heat Gain Through Windows With Venetian Blinds	Excess Solar Heat Gain Through Walls	Excess Solar Heat Gain Through Roof	Total Excess Solar Heat Gain (Cols. 2+3+4)	Total Heat Gain[a]
1	2	3	4	5	6
6 a.m.	5,460	.....	.....	5,460	101,560
7 a.m.	15,440	.....	.....	15,440	111,540
8 a.m.	20,160	435	.....	20,595	116,695
9 a.m.	21,630	4,350	.....	25,980	122,080
10 a.m.	21,630	6,310	.....	27,940	124,040
11 a.m.	18,060	6,090	17,550	41,700	137,800
12 M	15,540	6,310	50,700	72,550	168,650
1 p.m.	19,860	5,660	76,050	101,570	197,670
2 p.m.	26,850	4,790	97,500	129,140	225,240
3 p.m.	29,280	6,060	109,200	144,540	240,640
4 p.m.	28,710	7,460	113,100	149,270	245,370
5 p.m.	22,050	7,850	109,200	139,100	235,200
6 p.m.	7,800	8,270	97,500	113,570	209,670

[a] Column 6 was obtained by adding to the values in column 5 the heat gain of 96,100 Btu per hr obtained in the first part of the solution to Example 4.

Solar Heat Through Attic Floor

When there are windows in different walls of an attic, natural cross ventilation is obtained. In such a case, it is difficult to figure the actual excess solar heat gain through the floor of the attic to the rooms below. Although the solar heat gain through the roof itself can be figured, there is no accurate way of predicting how much of this heat will be carried away by the cross currents of air flowing through the attic and how much will be left to pass through to the ceiling of the room below. For this reason, past experience with attic temperatures is as good a guide for calculating the excess solar heat gain through the ceiling of rooms situated below such attics as any other method.

By keeping the attic windows open, the temperature in an attic can probably be held down to a point approximately 20 F above the outdoor temperature. Therefore, the excess solar heat gain through the ceiling of the room below should be figured for a 20 deg solar temperature difference. As heretofore, the excess solar heat gain is added to the conduction heat gain computed for a 15 deg difference. When the temperature of the attic is assumed, as in this case, only the over-all coefficient for the floor of the attic and the ceiling below should be used—the coefficient for the roof above should be ignored.

The data in Tables 1 to 8 can be used for the summer season in latitudes south of the equator by interchanging the north and south directions throughout the tables; the east and west directions remain the same. For example, in the southern hemisphere, for a north wall, read the column in Tables 1 to 8 marked "S"; for a northwest wall, read the column marked "SW"; and for a northeast wall, read the column marked "SE."

[To be continued]

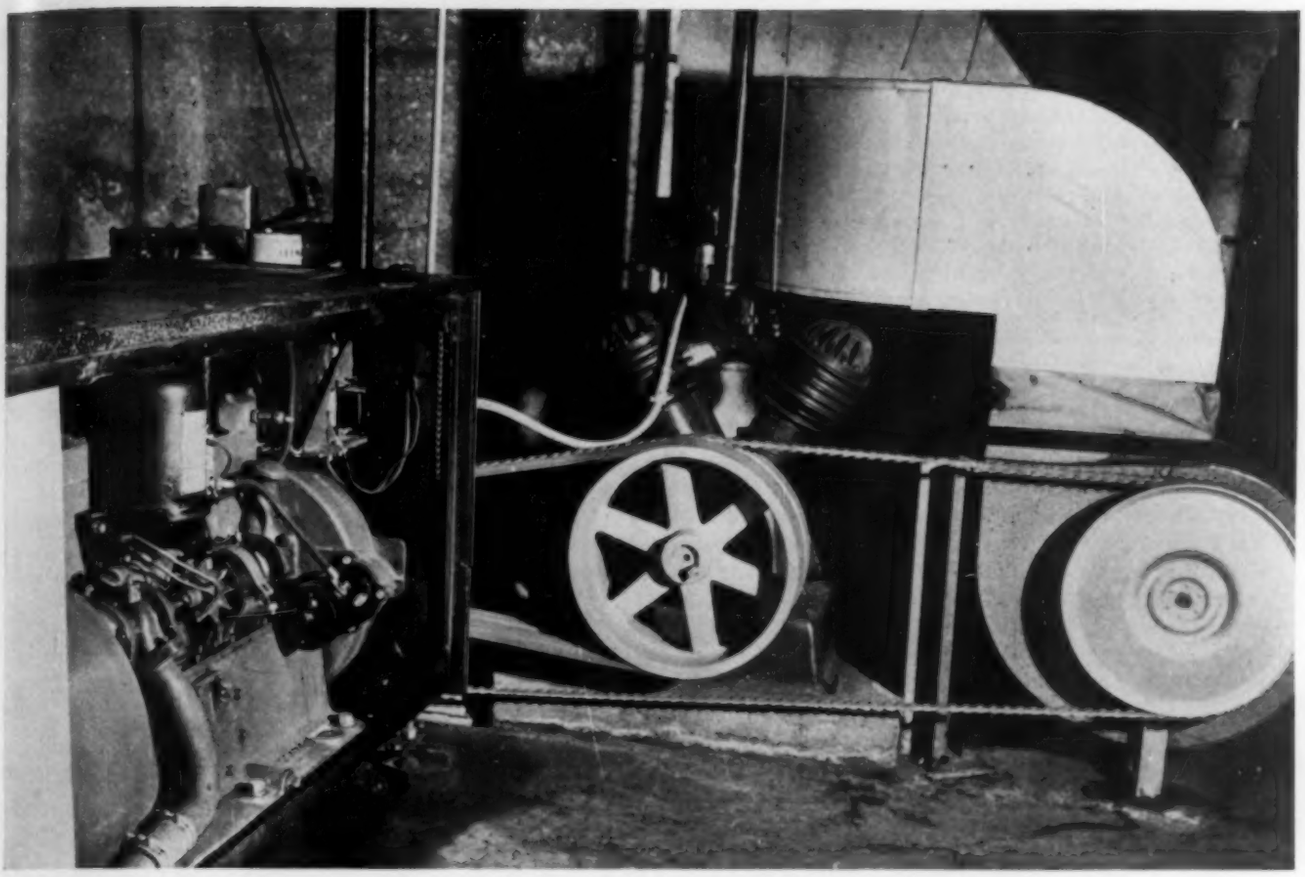


Fig. 1—Gas engine driven storage installation for air conditioning a dance hall with a load of 100 tons. Fifteen tons of ice are produced in 24 hours at a cost of approximately 25 cents per ton of ice so accumulated.

Storing Refrigeration for Air Conditioning

Offers Possibilities in Many Cases, Should Be Investigated

THE refrigeration storage principle offers a number of possible advantages in air conditioning, should be investigated in many cases. According to W. F. Friend,[1] it is the most feasible single means of reducing the maximum demand on its power supply source by an air conditioning system. It is possible not only to reduce the magnitude of the maximum load, but also to control the time of its occurrence. Savings are greatest on electrical rates with high demand charges. In localities where lower electrical rates are in effect for off-peak operation, storage should be considered. The reduced capacity required and improved load factor are also of benefit with the use of gas or diesel engines for driving compressors, where their fuels compete with electricity (Fig. 1).

When cooling towers are used for the condensing water supply, the storage type of system has an advantage in some saving in power consumption due to the lower average outdoor wet bulb temperatures prevailing at night.

An important application of the refrigeration storage principle is to increase the capacity of an existing air conditioning system if changes in the load make this necessary. At the central plant of Bayuk Cigars, Inc., Philadelphia, for instance, an alteration in the manufacturing process added a heat load of 42 kw or 2390 Btu per min. By operating the existing 120 ton centrifugal compressor during off-peak periods and storing chilled water in a tank on the roof, the additional refrigerating effect was obtained economically.[2]

Storing refrigerating effect generally means that the

[1] "Refrigeration Storage Applied to Air Conditioning," by W. F. Friend, presented at the annual meeting of the A. S. R. E., January, 1938, and published in *Refrigerating Engineering*, February, March, and April, 1938.

[2] "Increased Air Conditioning Capacity with Economy," by R. D. Touton, HEATING, PIPING AND AIR CONDITIONING, July, 1934.

The advantages — and disadvantages — of employing the refrigeration storage principle in air conditioning are briefly reviewed here. In many cases, it offers possibilities in reducing air conditioning costs, has been said to be "the most feasible means of reducing the maximum demand on its power supply source by an air conditioning system." . . . Storage of chilled water or accumulation of ice on submerged evaporator coils are thought to be the two methods of greatest promise in air conditioning

capacity of the refrigeration plant is reduced. With means for accumulating "cold," a small compressor running 22 to 24 hours a day can build up a reserve that is drawn on to meet the load on the air conditioning plant whenever this load occurs. In this way, the investment in the refrigeration equipment is reduced.

From the saving must of course be subtracted the cost of providing the means for storage. In many cases, the investment cost for the storage job may be greater than for a conventional system. The space required for the storage tank is a disadvantage. In existing buildings, this may present great difficulty. The question of attendance during night hours may be a problem with large jobs; on smaller installations, it has been found satisfactory to depend on automatic operation.

Required refrigeration capacity is not reduced below that for a conventional system when operation is contemplated exclusively at night to benefit from off-peak rates. In such cases, the storage plant may actually be larger than normal, if the night operating hours are substantially less than the daytime schedule for air conditioning service, or if ice storage is adopted with consequent lowered evaporator temperature and with resulting curtailed compressor capacity.

Saving in Investment

The saving in initial cost of the refrigeration equipment is the greatest when the hours of use of the air conditioning system are the shortest; extreme cases might be dance halls, meeting rooms, churches, mortuaries, legitimate theaters, restaurants doing the majority of their business at one meal, etc.

As an example of the extent to which the capacity and cost of the refrigerating plant might be reduced, a recent analysis made by Carl F. Boester is of interest. He assumes an installation with a maximum air conditioning load of 40 tons, that the system operates six hours per day, and that the average daily usage or load factor is 65 per cent. Using a storage system, a 10 ton compressor will handle the load.

For the storage system, the estimated cost of the refrigeration equipment is $3940. This does not include contractor's mark-up for overhead, profit or service maintenance, and does not include the air distribution system, duct work, grilles, etc., as these would be the same whether or not storage was used. The $3940 figure is intended to include one compressor with motor, drive, base, and pressure controls for use with an evaporative condenser, the compressor to be operated at a speed to produce 120,000 Btu per hr refrigeration at 20 deg under comparable condenser efficiencies; one 10 ton evaporative condenser; one 40 ton air conditioning unit with four row water coil; one chilled water pump with motor; one 10 ton storage unit with tank and evaporator, erected; and the labor cost for setting up and connecting the condenser, compressor, and air conditioning unit.

On the same basis the cost of a direct expansion plant without using refrigeration storage is estimated as $5375. This figure is intended to include two compressors operating to produce 15 tons of refrigeration each at a 45 deg evaporator temperature; one compressor having a capacity of 10 tons; one 40 ton evaporative condenser; one 40 ton direct expansion air conditioning unit, with four row direct expansion coil; controls for providing various capacities of refrigeration under partial load requirements; and labor and piping for installing compressors, evaporative condenser, air conditioning unit, and controls.

The net saving in this example by using the storage principle is therefore estimated at $1435.

Methods of Storing Refrigeration

There are various methods of storing refrigeration that have been successfully used on a commercial basis. As outlined by W. F. Friend,[1] they are:

Chilled water or brine, held in steel or concrete tanks.
Ice cakes, kept in dry storage and later charged into water filled tanks.
Ice layer, formed on evaporator coils submerged in water or eutectic solution tanks.
Eutectic brine ice, usually in sealed containers.
Solvent liquid in tanks, for absorbing refrigerant vapor.
Storage of steam in a steam accumulator for use with refrigeration systems of the steam jet and absorption types.

According to Friend, of these and other possible (but as yet not practicable) methods, storage of chilled water or formation of ice on submerged evaporator coils show the greatest promise for air conditioning.

Storage System for a Restaurant

Charles S. Leopold described in 1933[2] a storage system for cooling a 400 seat restaurant which serves lunch only. Due to the economies of the storage effect it was possible to use 40 F water in the washer, which would have necessitated approximately a 30 F evaporation temperature in a full capacity system. The actual storage installation makes use of a methyl chloride compressor of 8.65 tons, with an evaporator temperature running around 15 F for a large portion of the cycle and decreasing somewhat as the ice thickness builds up.

The tank occupies a space 6' x 6' x 20' and is so arranged that the system can store six tons of ice in 18 hours, the six hour margin of safety (24 hours minus 18 hours) being allowed for certain engineering contingencies which had never been adequately investigated, and to permit of occasional stopping of the compressor to prevent coincidence of electrical load peaks. The plant is entirely automatic in operation, requiring no more attention than a full automatic refrigerating machine attached to a water storage tank, and less attention than a direct operating unit. The ice is, of course, not handled in any way. It forms on coils in definite channels during

[2] "Reduces Air Conditioning Costs with Compressor Equipped Plant Storing Refrigerating Effect," by Charles S. Leopold, HEATING, PIPING AND AIR CONDITIONING, June, 1933.

the period it is built up and is melted down by the water returning from the dehumidifier.

A ton of ice can be manufactured more cheaply by accumulating it on an evaporator than it can be made by freezing it in cans suspended in a brine tank. The thickness to which the ice is frozen, and the fact that the evaporator operates at a higher suction temperature, means a lower over-all cost for power per ton of ice produced.

A Storage System Design

A design employed by Carl F. Boester for a number of storage jobs uses a coil constructed of ⅝ in. copper tubing placed on 3½ in. centers, supported on aluminum fins spaced 10 to 12 in. apart. A portion of such an evaporator section is illustrated in Fig. 2. These sections are set in the accumulator tank so that when the ice is formed it appears as if blocks of ice were suspended side by side in a tank of water. The number of lineal feet of tubing used per ton of ice accumulated depends on the cost for power. Where power costs are high, it is desirable to use more surface per ton of ice to be accumulated. Where more surface is required because of higher power costs, the centers on which the tubes are placed would naturally be closer.

The length and number of passes is determined by the pressure drop across the coil and by the physical dimensions of a section of the evaporator coil. The evaporator sections are not made in standard sizes because space requirements where storage refrigeration is used vary greatly. Thus some storage tanks, because of the peculiarities of the particular space in which they must be located, are considerably greater in height than in width, etc. The evaporator sections must, of course, conform to the dimensions of the tank.

Copper tubing is used because it is smooth, requires less refrigerant and causes less pressure drop. The refrigerant is fed dry through the evaporator coil and is controlled by thermostatic expansion valves. Due to the high cubic content of the entire length of evaporator coil, it is not economical to operate the coil flooded or semi-flooded. If it were operated flooded, the quantity of refrigerant required would add materially to the cost of the system and partially defeat the advantage gained by low first cost, if "Freon" were the refrigerant.

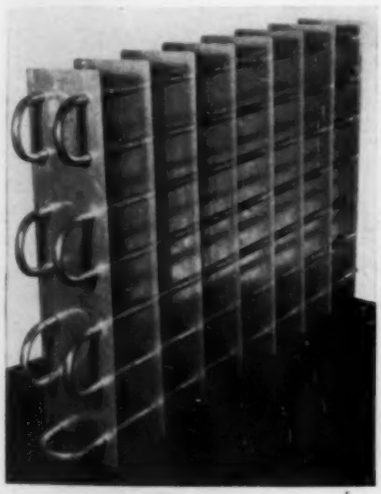


Fig. 2 — Portion of one type of evaporator section on which ice is accumulated for storage of refrigeration for air conditioning

The method of feeding the passes in the evaporator section is determined in part by the velocity of the refrigerant through the coil and by the pressure drop across the coil due to the length and the number of passes.

Ice is lighter than water and therefore has a tendency to float, and, as the evaporator sections are suspended in water, there is no weight on the coils. It is necessary to anchor the evaporator to the bottom or sides of the tank in order to prevent its floating too much, shifting and causing a break in refrigerant lines.

The evaporators are made up into sections so that when they are solidly accumulated with ice there is sufficient square feet of ice surface for the water in the storage tank to flow over and around, thereby permitting the chilling of the water and the melting of the ice. The evaporator sections are placed on close centers in the tank, yet far enough apart so that the water between sections does not freeze up and prevent the proper flow of water. It is desirable that a definite volume of water at a constant temperature be supplied to the air cooling coils if control of the temperature of the air being cooled is expected. Otherwise it would be necessary to provide over-sized air cooling coils because of the uncertain water temperature, and this would be objectionable because it would increase first cost.

Control of Water Temperature

Most of these storage units are designed to insure that 33 F water leaves the unit at all times, but this temperature is not desirable in the air stream cooling coils, and a three way proportioning valve is used to blend a quantity of the return water from the cooling coils with the chilled water being withdrawn from the storage unit. This mixing valve assures a constant temperature water to the cooling coils at all times regardless of load conditions. For instance, if, on a particular application, 45 F water is required in the air stream cooling coils, then the mixing valve is so adjusted to provide this temperature water.

The type of evaporator design described insures a great quantity of usable surface at low first cost, and this means that it is possible to maintain suction temperatures averaging around 25 to 28 F, so that there is little power penalty. A ton of ice can be produced in a space of 45 to 50 cu ft.

Bypass Around Valve

A bypass around the proportioning valve from the return water line of the coils is usually provided. It is sometimes desirable to run the pump constantly circulating the water in the tank in order to pull down the water temperature in the tank more rapidly when there is no ice on the evaporator coils. Generally this is necessary only when the system is started up at the beginning of each season, for the storage units are so designed that at no time is the ice entirely washed off the evaporator coils. If the water temperature is allowed to rise, the suction pressure rises due to the excessive amount of coil surface in the tank. This also raises the head pressure and causes the high pressure cutout to stop the machine. It then becomes necessary either to use a suction pressure regulator or to choke back on the suction stop valve on the compressor in order to prevent this overloading. This in turn cuts down the compressor capacity, which is not desirable because it would then become necessary to run the compressor for more than a 24 hour period in order to accumulate the quantity of ice necessary for use during the actual hours of cooling operation.

Weld Design

for High Pressure, High Temperature Piping

By J. H. Deppeler*

For the successful arc welding of high pressure, high temperature piping it is essential that the weld design is correct, a suitable welding technic is evolved, the welding procedure and the welding operators are properly qualified, and that the proper electrodes are selected. The first of these factors is briefly discussed here

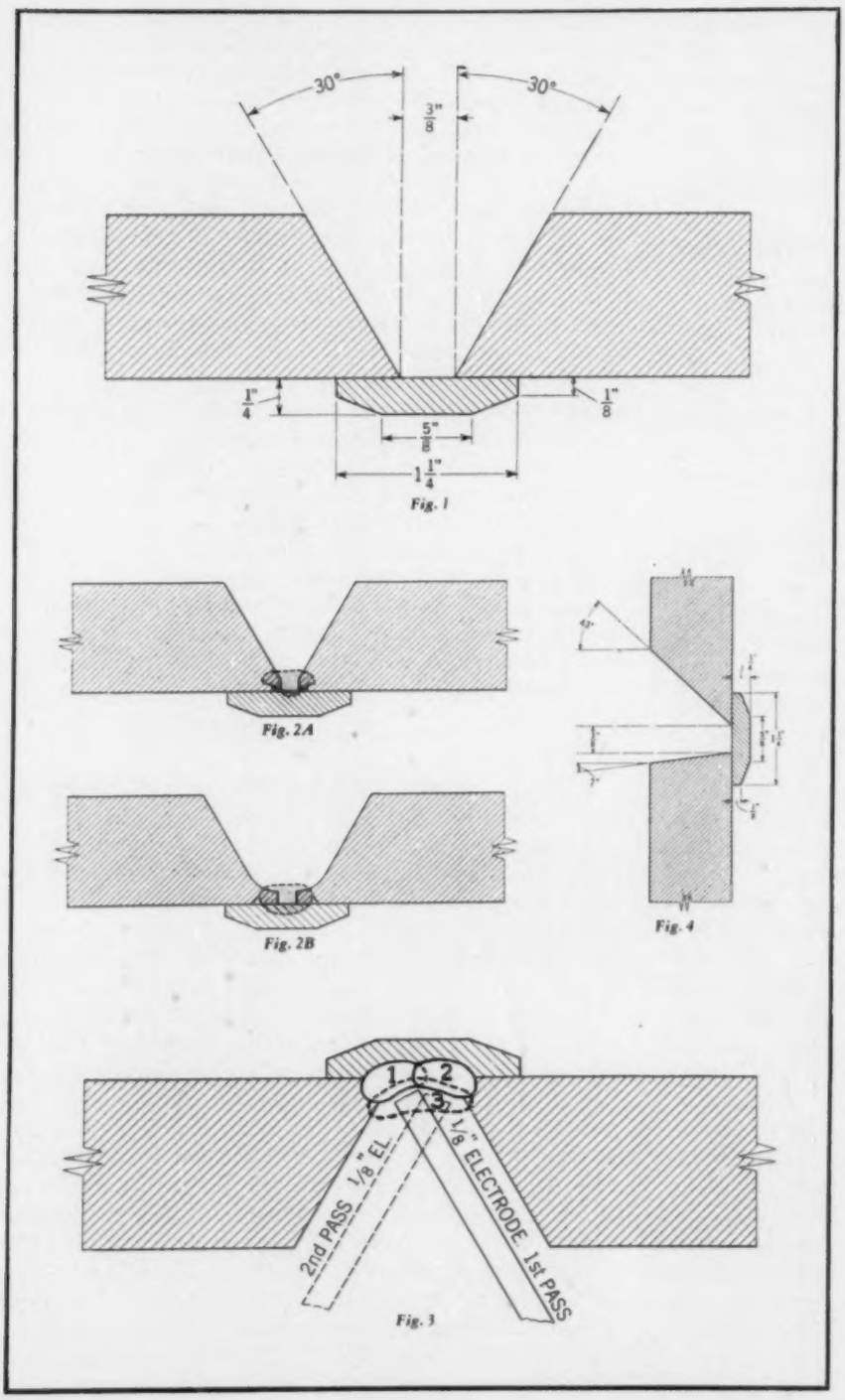


Fig. 1

Fig. 2A

Fig. 2B

Fig. 4

Fig. 3

A NUMBER of different types of welded joints for pipe lines have been designed and employed in installations to date. Some of these designs call for recessed and raised bead backing strips or chill rings; others involve upsetting of the pipe ends to permit recessing of the chill ring in the pipe wall without reducing the cross section at the weld; and still others require machining of pipe edges to form "U" grooves and bevels with lips of varying dimensions. Good welds may be produced and entirely satisfactory results have been obtained with any of these designs. All things considered, however, the primary requisite of any weld design is weldability and, as a general rule, the more simplified the design the easier it becomes for operators to produce sound and satisfactory welds.

From this standpoint, the basic design, shown in Fig. 1, possesses many advantages. This design is intended for welds in plain carbon steel or carbon molybdenum steel pipe 2½ in. or more in diameter in the horizontal rolled, or "A" position, and the horizontal fixed, or "B" position. It calls for a 30 deg straight bevel of each pipe edge with no lip, with a plain chill ring of proper dimensions, and with a ⅜ in. gap at the root of the "V."

The chill ring is an essential feature of all pipe weld designs. Its purpose is to prevent formation of icicles and weld spatter inside the pipe and to provide complete fusion at the root of the weld. Chill rings are generally made of the same material as the pipe being welded. Since pipe walls may vary slightly in thickness and pipe is seldom perfectly round, it is frequently necessary, when using chill rings, to machine the pipe ends in order to provide true round inside contours and assure a good fit for the rings.

The ⅜ in. gap at the root of the "V" is recommended because, while perfectly satisfactory welds can be made with smaller gaps, the larger space permits better manipulation of the electrode during welding, makes easier the laying down of the first few beads, which are usually the most difficult, and results in better welds.

The chill ring is not recessed in the pipe wall in this design because in an investigation made for The Detroit Edison Co. at the University of Michigan, it was definitely determined that the slight reduction in inside diameter

[Concluded on p. 400]

*Chief Engineer, Metal & Thermit Corp.

Refrigeration Trouble-Shooting

By W. A. Stains*

THE refrigerant "Freon" has a propensity to escape through minute leaks, and quite often after charging the system with the required amount one finds that low suction pressures exist. Regardless of the type of refrigerant, if the system for any reason is low on refrigerant the suction pressure will be too low. This is sometimes manifested by only part of the evaporator operating at proper temperature, the rest being warm.

Where there are several sets of coils one above the other, fed by expansion valves some of which are at a higher elevation than others (Fig. 1), the upper valves may apparently not be feeding sufficient liquid and the top coils will suffer. This is because the large volume of gas which may form in the liquid lines will rise above the liquid and be vented through the top expansion valves. As an expansion valve is selected for passing solid liquid and the volume of gas is many times the volume of liquid, it is evident that the same orifice opening designed for a volume of liquid can't possibly pass the resultant volume of gas. Even if it could, the heat absorption of the gas is not comparable to the liquid because the greater heat is absorbed in the transition from liquid to gas.

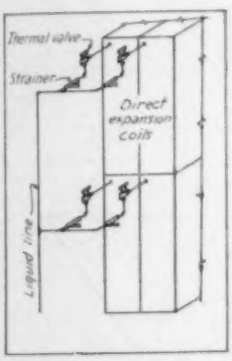


Fig. 1—On an installation like this, gas formed in the liquid lines may rise above the liquid and be vented through the top expansion valves. Performance of the top coils consequently will suffer

In many cases where several unit coolers are fed from a common liquid line with the branches teeing off from the liquid main, (Fig. 2), the refrigerant shortage is manifested by both low suction pressure at the compressor and failure of the units to perform satisfactorily. Those units at the end of the liquid main may function nicely and the others fail entirely. This is because the flow of liquid through the main, due to its velocity, will go right by a tee branch opening, especially if the branch is taken off the main horizontally or off the top. The gas in the liquid line will take the course of the branch while the liquid flows straight ahead. Additional refrigerant must be added to the system to correct this condition.

Pipe Sizing Most Important

The matter of pipe sizing is tremendously important, especially for liquid lines. Failure of many systems to carry the expected loads can be attributed to too small liquid lines, particularly with systems having long runs from compressor to evaporator. Resistance to flow between the receiver and evaporator is equivalent to pressure reduction. When the pressure on the refrigerant is reduced, a certain amount of the refrigerant changes from liquid to gas in the liquid lines and the same condition is encountered as when there is an insufficient charge in the system. Suction pressures drop and often the connection between the expansion valve and the evaporator becomes frosted. The liquid line at the evaporator usually is colder than ordinarily due to the partial expansion of the liquid in the line and this is especially true where there is no subcooling of the liquid refrigerant in the condenser.

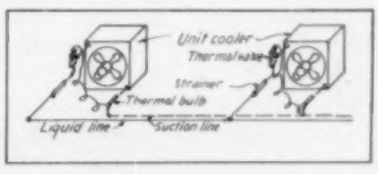
There is no real cure for too small liquid lines except to tear them out and increase their size. The best method is to furnish ample sized lines in the first place as the extra cost of one or two sizes larger pipe is a relatively small part of the total installation cost. Some attempts have been made to install liquid subcooling devices to overcome flashing of liquid into gas in the liquid lines due to pressure release, but this method is ineffective unless very carefully worked out. Where doubt exists as to whether there is gas in the liquid lines, sight glasses can be installed at the evaporator for observation.

The suction line should also receive attention. A small line between the evaporator and the compressor causes a pressure drop that, while the suction pressure at the evaporator may be as desired, penalizes the capacity of the compressor (which usually is selected to deliver rated tonnages at a suction pressure very close to the evaporator pressure). Several pounds difference between evaporator and compressor reduces compressor capacity and handicaps performance of the system.

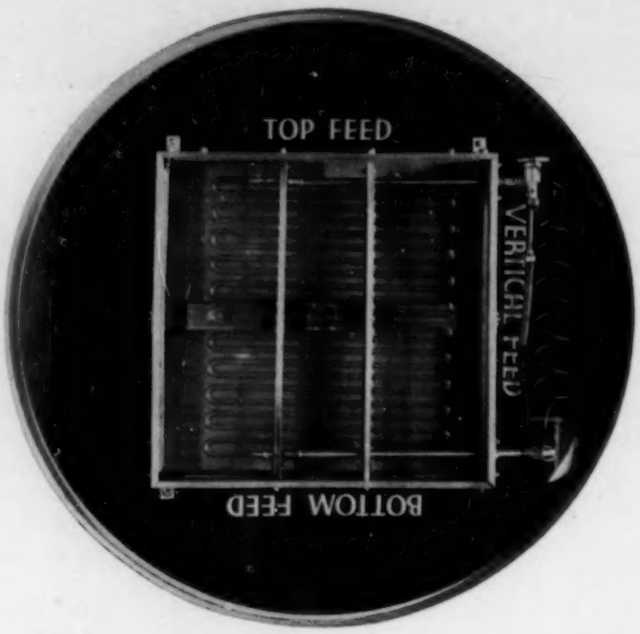
Restrictions in Piping

With welded or soldered lines it is essential that projections inside them be eliminated. On larger pipes the "grapes" from welding or solder lumps on copper pipe construction are not as likely to cause trouble as on smaller pipes. If a restriction does occur in a liquid line it will act as a fixed expansion valve and the pressure drop will cause gas to form with the same results as stated before. One of the most persistent points of trouble due to solder restricting or filling a line is in the ¼ in. O. D. copper equalizing line from the expansion valve to the evap-

Fig. 2—Where several unit coolers are fed from a common liquid line, units at the end may function nicely and the others fail to do so, because of the liquid going right by the branch openings to the various units



*Sales Engineer, Alco Valve Co., Inc.



GLASS EVAPORATOR

In an attempt to analyze the action of a refrigerant after passing through the expansion valve and its effect upon the main controlling element, the thermal bulb, pressure gages, thermocouples and other instruments failed to provide the full story. Only visual observation was the answer to augment the results of the instruments. The glass evaporator illustrated was then designed and assembled, and connected to a small "Freon" compressor unit. Load conditions were produced by directing a blast of air over the glass coil. . . . The glass evaporator is designed to permit top, bottom or vertical feed of refrigerant into the coil under identical load conditions. Much has been learned from its observation as to the causes producing certain results that were formerly not understood. Among these are the separation of gas and liquid, oil travel with refrigerant, impingement, etc.

orator when external equalizing valves are used.

A common cause of restriction trouble in liquid lines is the use of shut-off valves or solenoid liquid stop valves that have high pressure drops across the valve. These valves should be selected on the basis of very low pressure drops across them, best practice being to allow not more than 2 lb drop.

King valves on receivers are quite often of comparatively small size. I recall one instance when a "Freon" unit designed for 40 tons duty had but a 1 in. king valve. The principal trouble with this arrangement is that the installation engineer may accept this size as a large enough pipe size, whereas for 40 tons capacity no less than a 1½ or 2 in. liquid line should be used for short runs, and a larger size should be used where runs are of considerable length.

Another cause of restriction in liquid lines—and one often overlooked—is a scale trap, filter or device for catching dirt or scale which becomes clogged.

[Mr. Stains' next article will consider evaporators above condensing units and use of heat interchangers.]

Weld Design—

[Concluded from p. 398]

of the pipe caused by an unrecessed chill ring has no appreciable effect on steam flow. The additional expense that recessing entails, as well as any need for upsetting of pipe ends to compensate for reduced wall thickness, are thus avoided.

For several reasons the design is an improvement over those calling for "U" grooves or "V's" with pipe ends beveled to provide lips at the root of the "V." For example, when depositing the first layer of weld metal at the root of a "U" groove, or in a "V" with lips, it is necessary for the welder not only to melt away the edge of each lip but also, and at the same time, to penetrate into both pipe wall and chill ring. This is not always a simple matter, particularly where the gap is too narrow to permit easy manipulation of the electrode, or in welding on the underside of a pipe in the horizontal position, where it is difficult for the welder to watch the arc. Actually, the results of many tests show, for example, that other things being equal, more defective welds can be traced directly to this particular condition than to any other source. Fig. 2 indicates the difficulty of producing sound deposits with welds of such design and shows how readily slag is entrapped and cold shuts occur. Fig. 3, on the other hand, shows how, with straight beveled pipe ends and a gap of ample width, the welder can easily produce sound, clean deposits at the root by penetrating slightly into the chill ring and washing lightly up the pipe wall, first at one side of the root and then the other.

For welds in the vertical, or "C" position, the design is modified, as shown in Fig. 4. The bevel of the lower pipe edge is reduced to approximately 7 deg while that of the upper pipe edge is increased to about 45 deg. The gap remains at ⅜ in. This design for vertical welds provides a nearly horizontal shelf for the operator to build on and makes every bead deposited practically a fillet weld. The welding technic thus becomes simply that of continuous fillet welding.

It is recommended that the design for horizontal welds shown in Fig. 1 be employed for all welding on pipe in the horizontal position, as well as on pipe up to an angle of 30 deg from the horizontal. Where the pipe to be welded is inclined at an angle of more than 30 deg from the horizontal, however, the design for vertical welds, shown in Fig. 4, should be used.

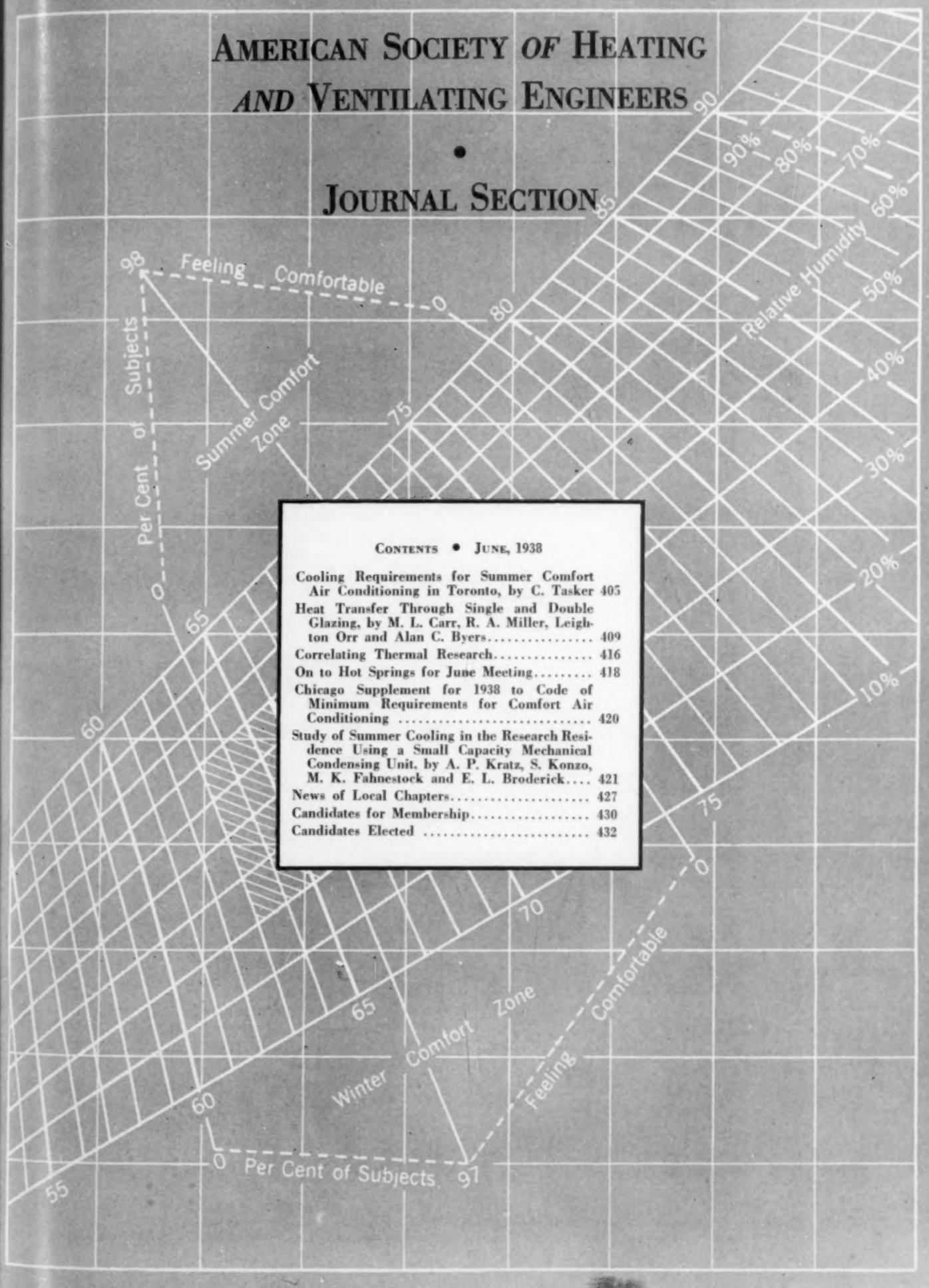
For welds in the smaller diameters of both carbon molybdenum and mild steel pipe, a 60 deg included V design of weld, similar to that shown in Fig. 1 but with a $\frac{3}{16}$ in. to ¼ in. gap and a ⅛ in. thick chill ring, may be employed in all positions.

Heating · Piping
and Air Conditioning

AMERICAN SOCIETY *OF* HEATING *AND* VENTILATING ENGINEERS

•

JOURNAL SECTION



CONTENTS • JUNE, 1938

Cooling Requirements for Summer Comfort Air Conditioning in Toronto, by C. Tasker 405
Heat Transfer Through Single and Double Glazing, by M. L. Carr, R. A. Miller, Leighton Orr and Alan C. Byers 409
Correlating Thermal Research 416
On to Hot Springs for June Meeting 418
Chicago Supplement for 1938 to Code of Minimum Requirements for Comfort Air Conditioning 420
Study of Summer Cooling in the Research Residence Using a Small Capacity Mechanical Condensing Unit, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick 421
News of Local Chapters 427
Candidates for Membership 430
Candidates Elected 432

OFFICERS AND COUNCIL—1938

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

51 Madison Ave., New York, N. Y.

OFFICERS

President.....E. HOLT GURNEY
First Vice-President.....J. F. McINTIRE
Second Vice-President.....F. E. GIESECKE
Treasurer.....A. J. OFFNER
Secretary.....A. V. HUTCHINSON
Technical Secretary.....JOHN JAMES

COUNCIL

E. HOLT GURNEY, *Chairman* — J. F. McINTIRE, *Vice-Chairman*

Three Years

N. D. ADAMS — A. P. KRATZ, G. L. WIGGS — J. H. WALKER

Two Years

J. J. AEBERLY — M. C. BEMAN, W. A. RUSSELL — E. O. EASTWOOD

One Year

R. C. BOLSINGER, W. L. FLEISHER — D. S. BOYDEN — S. H. DOWNS, C. M. HUMPHREYS

COMMITTEES OF THE COUNCIL

Executive: D. S. BOYDEN, *Chairman*
W. L. FLEISHER — F. E. GIESECKE

Finance: J. F. McINTIRE, *Chairman*
S. H. DOWNS — E. O. EASTWOOD

Meetings: J. J. AEBERLY, *Chairman*
N. D. ADAMS — A. P. KRATZ

Membership: R. C. BOLSINGER, *Chairman*
W. A. RUSSELL — G. L. WIGGS

Standards Committee: C. M. HUMPHREYS, *Chairman*
M. C. BEMAN — J. H. WALKER

ADVISORY COUNCIL

D. S. Boyden, *Chairman;* Homer Addams, R. P. Bolton, W. H. Carrier, S. E. Dibble, W. H. Driscoll, H. P. Gant, John F. Hale, L. A. Harding, H. M. Hart, C. V. Haynes, E. Vernon Hill, J. D. Hoffman, John Howatt, W. T. Jones, D. D. Kimball, G. L. Larson, S. R. Lewis, Thornton Lewis, J. I. Lyle, F. B. Rowley, F. R. Still and A. C. Willard.

COOPERATING COMMITTEE

A. S. H. V. E. Representative on *A. S. A. Sectional Committee on Standardization of a Scheme for Identification of Piping Systems (A-13):* E. E. Ashley.

SPECIAL COMMITTEES

Committee on Admission and Advancement: T. H. Urdahl, *Chairman (one year);* M. F. Blankin *(two years),* and A. A. Adler *(three years).*

Publication Committee: F. C. McIntosh, *Chairman (one year);* W. M. Sawdon *(two years),* and W. H. Severns *(three years).*

Guide Publication Committee: Albert Buenger, *Chairman;* S. H. Downs, C. H. B. Hotchkiss, S. S. Sanford, G. L. Tuve, and E. N. McDonnell, *Advisory.*

Committee on Constitution and By-Laws: John Howatt, *Chairman;* O. W. Ott, and W. E. Stark.

F. Paul Anderson Award Committee: J. F. McIntire, *Chairman;* E. O. Eastwood, W. T. Jones, A. P. Kratz, and G. L. Larson.

Speakers Bureau Committee: W. A. Russell, *Chairman;* L. T. Avery, B. C. Candee, O. J. Prentice, and Dr. B. M. Woods.

A. S. H. V. E.—I. E. S. Committee on Lighting in Air Conditioning: H. M. Sharp, *Chairman;* E. E. Ashley, A. A. Brainerd, A. D. Cameron, F. H. Faust, W. F. Friend, W. E. Stark, and Walter Sturrock.

Committee on A. S. H. V. E. Code for Testing Stoker Fired Heating Boilers: C. E. Bronson, *Chairman;* L. A. Harding, H. M. Hart, F. C. Houghten, A. J. Johnson, J. F. McIntire, D. W. Nelson, Percy Nicholls, R. A. Sherman, and E. C. Webb.

Fan Test Code Committee: S. R. Lewis, *Chairman;* J. J. Aeberly, S. H. Downs, W. A. Rowe, A. E. Stacey, Jr., and Ernest Szekely.

Committee on Engineering in Its Relation to Public Health: J. J. Aeberly, *Chairman;* W. H. Carrier, Philip Drinker, Dr. E. V. Hill, John Howatt, Dr. A. C. Willard and Dr. B. M. Woods.

Educational Publicity Committee: J. M. Frank, *Chairman;* C. K. Foster, H. M. Hart, E. P. Heckel, E. N. McDonnell, and J. R. Vernon.

NOMINATING COMMITTEE

Chapter	*Representative*	*Alternate*
Atlanta	E. W. Klein	C. T. Baker
Cincinnati	I. B. Helburn	H. E. Sproull
Golden Gate	B. M. Woods	
Illinois	J. R. Vernon	Tom Brown
Iowa-Nebraska	A. L. Walters	M. J. Stevenson
Kansas City	A. H. Sluss	A. L. Maillard
Manitoba	William Glass	R. L. Kent
Massachusetts	C. P. Yaglou	W. T. Jones
Michigan	F. J. Feely	F. J. Linsenmeyer
Western Michigan	W. W. Bradfield	L. G. Miller
Minnesota	R. E. Backstrom	M. S. Wunderlich
Montreal	L. H. Laffoley	G. L. Ballantyne
New York	W. E. Heibel	E. J. Ritchie
Western New York	B. C. Candee	J. J. Landers
Northern Ohio	L. T. Avery	G. L. Tuve
Oklahoma	E. F. Dawson	E. W. Gray
Ontario	H. R. Roth	C. Tasker
Pacific Northwest	C. W. May	L. Bouillon
Philadelphia	H. H. Erickson	H. H. Mather
Pittsburgh	J. F. Collins, Jr.	R. A. Miller
St. Louis	G. W. F. Myers	R. J. Tenkonohy
Southern California	E. H. Kendall	R. L. Gifford
Texas	W. H. Badgett	I. E. Rowe
Washington, D. C.	M. D. Kiczales	T. H. Urdahl
Wisconsin	C. H. Randolph	E. Szekely

[The material in this Section is prepared by the Publication Committee of the Society and nothing that appears in it may be reprinted, either wholly or in part, without permission. The use of the Society's name for advertising promotions is never authorized.]

COMMITTEE ON RESEARCH

W. L. FLEISHER, *Chairman*

J. H. WALKER, *Vice-Chairman* F. C. HOUGHTEN, *Director*

A. C. FIELDNER, *Ex-Officio Member*

Executive Committee

W. L. FLEISHER

W. A. DANIELSON C. TASKER J. H. WALKER C.-E. A. WINSLOW

One Year

W. A. DANIELSON C. E. LEWIS D. W. NELSON

C. TASKER C.-E. A. WINSLOW

Two Years

H. E. ADAMS A. E. STACEY, JR. G. L. TUVE

J. H. VAN ALSBURG J. H. WALKER

Three Years

M. K. FAHNESTOCK W. L. FLEISHER R. J. TENKONOHY

T. H. URDAHL B. M. WOODS

Technical Advisory Committees

1. *Relations of Air Conditions to Human Health and Comfort.* W. L. Fleisher*, *General Chairman*

   1a. *Sensations of Comfort:* C. Tasker*, *Chairman;* Thomas Chester, F. E. Giesecke, Elliott Harrington, R. E. Keyes, Dr. W. J. McConnell, A. B. Newton, J. R. Parsons, C. P. Yaglou.

   1b. *Physiological Reactions:* C.-E. A. Winslow*, *Chairman;* Dr. E. F. DuBois, Dr. R. W. Keeton, André Missenard, Dr. R. R. Sayers.

   1c. *Treatment of Disease:* Dr. T. L. Hazlett, *Chairman;* Dr. C. J. Barone, Dr. B. Z. Cashman, Dr. M. B. Ferderber, Dr. E. V. Hill, C. S. Leopold, Dr. C. D. Selby, Dr. W. O. Sherman, Dr. A. W. Sherrill, Dr. H. F. Smith, R. J. Tenkonohy*, Dr. B. L. Vosburgh.

   1d. *Climate and Season:* J. H. Walker*, *Chairman;* Dr. H. A. Abramson, O. W. Armspach, Ellsworth Huntington, Dr. C. A. Mills, André Missenard, T. H. Urdahl*, E. L. Weber.

   1e. *Air Conditioning in Industry:* A. E. Stacey, Jr.*, *Chairman;* Philip Drinker, Dr. Leonard Greenburg, A. M. Kinney, J. W. Kreuttner, L. L. Lewis, C. P. McCord, P. A. McKittrick, Dr. R. R. Sayers.

2. *Air Cleaning and Atmospheric Impurities.* H. C. Murphy, *General Chairman*

   2a. *Mechanical Procedures:* Dr. Leonard Greenburg, *Chairman;* J. J. Bloomfield, W. H. Carrier, Theodore Hatch, C. E. Lewis*, A. L. Simison, W. O. Vedder.

   2b. *Electrical Procedures:* Dr. E. B. Phelps, *Chairman;* H. E. Adams*, R. D. Bennett, L. W. Chubb, L. R. Koller, G. W. Penney, W. F. Wells.

3. *Radiation and Comfort Winter and Summer and Effect of Varying Humidity on Radiant Heating and Cooling:* J. C. Fitts, *Chairman;* A. A. Adler, A. H. Barker, W. D. Fleming, R. P. James, Dr. C. A. Mills, D. W. Nelson*, W. R. Rhoton, W. W. Timmis, G. R. Wait, S. L. Warren, C.-E. A. Winslow*, C. F. Wood.

5. *Weather Design Conditions:* T. H. Urdahl*, *Chairman;* F. S. Cornell, John Everetts, Jr., E. W. Goodwin, A. C. Grant, J. H. Kincer, O. A. Kinzer, J. W. O'Neill, L. S. Ourusoff.

6. *Transportation Air Conditioning:* J. H. Van Alsburg*, *Chairman;* W. I. Cantley, T. R. Crowder, A. G. Dixon, C. C. Elmes, L. H. Laffoley, E. A. Russell.

7. *Radiation with Gravity Air Circulation:* M. K. Fahnestock*, *Chairman;* B. C. Benson, H. F. Hutzel, J. P. Magos, J. W. McElgin, J. F. McIntire, D. W. Nelson*, T. A. Novotney, W. A. Rowe.

8. *Heat Transfer of Finned Tubes with Forced Air Circulation:* G. L. Tuve*, *Chairman;* W. E. Heibel, H. F. Hutzel, R. F. Norris, R. H. Norris, C. H. Randolph, L. P. Saunders, C. F. Wood.

9. *Cooling Load in Summer Air Conditioning:* J. H. Walker*, *Chairman;* C. M. Ashley, John Everetts, Jr., F. H. Faust, A. E. Knapp, L. S. Morse, A. E. Stacey, Jr.*, R. M. Strikeleather.

10. *Solid Fuels:* W. A. Danielson*, *Chairman;* H. N. Eavenson, A. J. Johnson, Percy Nicholls, H. J. Rose, J. E. Schoen, L. E. Seeley, E. T. Selig, R. A. Sherman, R. Templeton Smith, C. Tasker*.

11. *Summer Air Conditioning for Residences:* M. K. Fahnestock*, *Chairman;* E. A. Brandt, John Everetts, Jr., Elliott Harrington, H. F. Hutzel, E. D. Milener, K. W. Miller, R. E. Robillard, F. G. Sedgwick, J. H. Walker*.

12. *Air Distribution and Air Friction:* J. H. Van Alsburg*, *Chairman;* S. H. Downs, M. K. Fahnestock*, F. J. Kurth, R. D. Madison, L. G. Miller, D. W. Nelson*, C. H. Randolph, Ernest Szekely, G. L. Tuve*.

13. *Heat Requirements of Buildings:* P. D. Close, *Chairman;* W. H. Badgett, E. F. Dawson, W. H. Driscoll, E. K. Campbell, H. M. Hart, H. H. Mather, F. B. Rowley, R. J. J. Tennant, J. H. Walker*.

14. *Air Conditioning Requirements of Glass:* M. L. Carr, *Chairman;* F. L. Bishop, A. N. Finn, S. O. Hall, E. H. Hobbie, R. A. Miller, F. W. Parkinson, Harold Perrine, W. C. Randall, L. T. Sherwood, J. T. Staples, C. Tasker*, G. B. Watkins, F. C. Weinert.

15. *Insulation:* W. A. Danielson*, *Chairman;* E. A. Allcut, H. C. Bates, Wharton Clay, H. C. Dickinson, J. D. Edwards, E. C. Lloyd, Paul McDermott, W. E. McMullen, R. T. Miller, E. R. Queer, T. S. Rogers, F. B. Rowley, W. S. Steele, R. J. Tenkonohy*, G. B. Wilkes.

16. *Sound Control:* J. S. Parkinson, *Chairman;* C. M. Ashley, G. F. Drake, A. M. Greene, Jr., A. L. Kimball, V. O. Knudsen, R. F. Norris, C. H. Randolph, J. P. Reis, W. R. Roop, A. E. Stacey, Jr.*, G. T. Stanton, F. R. Watson.

17. *Cooling Towers, Evaporative Condensers and Spray Ponds:* B. M. Woods*, *Chairman;* J. C. Albright, S. C. Coey, E. R. Goodrich, E. H. Hyde, E. H. Kendall, S. R. Lewis, O. W. Ott, E. H. Taze.

18. *Psychrometry:* F. R. Bichowsky, *Chairman;* C. A. Bulkeley, J. A. Goff, A. M. Greene, Jr., F. G. Keyes, D. W. Nelson*, W. M. Sawdon.

19. *Corrosion in Steam Systems:* A. R. Mumford, *Chairman;* H. E. Adams*, J. F. Barkley, W. H. Driscoll, T. J. Finnegan, R. M. Palmer, R. R. Seeber, F. N. Speller, C. M. Sterne.

20. *Corrosion in Air Conditioning Equipment:* A. E. Stacey, Jr.*, *Chairman;* A. F. L. Anderson, G. L. Cox, M. L. Diver, W. R. Heath, C. E. Lewis*, R. M. Palmer, F. N. Speller, C. M. Sterne, J. H. Young.

*Member of Committee on Research.

OFFICERS OF LOCAL CHAPTERS—1938

ATLANTA: *Organized,* 1937. *Headquarters,* Atlanta, Ga. *Meets,* First Tuesday. *President,* C. L. TEMPLIN, Bona Allen Bldg., *Secretary,* H. K. McCAIN, 1311 W. Peach Tree St., N. E.

CINCINNATI: *Organized,* 1932. *Headquarters,* Cincinnati, O. *Meets,* Second Tuesday in Month. *President,* O. W. MOTZ, 2524 Moundview Dr., Norwood, O. *Secretary,* R. E. KRAMIG, JR., 222 East 14th St.

GOLDEN GATE: *Organized,* 1937. *Headquarters,* San Francisco, Calif. *Meets,* First Tuesday. *President,* B. M. WOODS, Univ. of Calif., Berkeley, Calif. *Secretary,* G. J. CUMMINGS, 113 Tenth St., Oakland, Calif.

ILLINOIS: *Organized,* 1906. *Headquarters,* Chicago, Ill. *Meets,* Second Monday. *President,* J. R. VERNON, 1355 Washington Blvd. *Secretary,* M. W. BISHOP, 228 N. La Salle St.

IOWA - NEBRASKA: *Organized,* 1937. *Headquarters,* Omaha, Neb. *President,* A. L. WALTERS, 900 29th St., Des Moines, Ia. *Secretary,* W. R. WHITE, 4339 Larimore Ave., Omaha, Neb.

KANSAS CITY: *Organized,* 1917. *Headquarters,* Kansas City, Mo. *Meets,* Second Monday in Month. *President,* A. H. SLUSS, 827 Mississippi Ave., Lawrence, Kan. *Secretary,* GUSTAV NOTTBERG, 914 Campbell St.

MANITOBA: *Organized,* 1935. *Headquarters,* Winnipeg, Man. *Meets,* Fourth Thursday. *President,* D. F. MICHIE, 492 Wardlaw Ave. *Secretary,* E. J. ARGUE, Ste. 23, Estelle Apts.

MASSACHUSETTS: *Organized,* 1912. *Headquarters,* Boston, Mass. *Meets,* Third Tuesday in Month. *President,* JAMES HOLT, Massachusetts Institute of Technology, Cambridge, Mass. *Secretary,* H. C. MOORE, 69 Massachusetts Ave., Cambridge, Mass.

MICHIGAN: *Organized,* 1916. *Headquarters,* Detroit, Mich. *Meets,* First Monday after the 10th of the Month. *President,* F. J. FEELY, 950 Trombley Rd., Grosse Pointe Pk. *Secretary,* G. H. TUTTLE, 2000 Second Ave.

WESTERN MICHIGAN: *Organized,* 1931. *Headquarters,* Grand Rapids, Mich. *Meets,* Second Monday in Month. *President,* W. W. BRADFIELD, 341 Michigan Trust Bldg. *Secretary,* S. W. TODD, JR., 309 Paris, S. E.

MINNESOTA: *Organized,* 1918. *Headquarters,* Minneapolis, Minn. *Meets,* Second Monday in Month. *President,* R. E. BACKSTROM, Room 1981, First Natl. Bank Bldg., St. Paul, Minn. *Secretary,* F. C. WINTERER, 836 Juno St., St. Paul, Minn.

MONTREAL: *Organized,* 1936. *Headquarters,* Montreal, Que. *Meets,* Third Monday. *President,* G. L. WIGGS, University Tower. *Secretary,* C. W. JOHNSON, 630 Dorchester St., W.

NEW YORK: *Organized,* 1911. *Headquarters,* New York, N. Y. *Meets,* Third Monday in Month. *President,* W. E. HEIBEL, 11 West 42nd St. *Secretary,* T. W. REYNOLDS, 100 Pinecrest Dr., Hastings-on-Hudson, N. Y.

WESTERN NEW YORK: *Organized,* 1919. *Headquarters,* Buffalo, N. Y. *Meets,* Second Monday in Month. *President,* J. J. LANDERS, 701 Crosby Bldg. *Secretary,* W. R. HEATH, 119 Wingate Ave.

NORTHERN OHIO: *Organized,* 1916. *Headquarters,* Cleveland, O. *Meets,* Second Thursday in Month. *President,* PHILIP COHEN, 401 East Ohio Gas Bldg. *Secretary,* C. A. McKEEMAN, Case School of Applied Science.

OKLAHOMA: *Organized,* 1935. *Headquarters,* Oklahoma City, Okla. *Meets,* Second Monday. *President,* E. F. DAWSON, University of Oklahoma, Norman, Okla. *Secretary,* E. W. GRAY, Box 1498, Oklahoma City, Okla.

ONTARIO: *Organized,* 1922. *Headquarters,* Toronto, Ont. *Meets,* First Monday in Month. *President,* H. B. JENNEY, Royce and Lansdowne Aves. *Secretary,* H. R. ROTH, 57 Bloor St., W.

PACIFIC NORTHWEST: *Organized,* 1928. *Headquarters,* Seattle, Wash. *Meets,* Second Tuesday in Month. *President,* W. W. Cox, 326 Columbia St. *Secretary,* M. N. MUSGRAVE, 314 Ninth Ave., N.

PHILADELPHIA: *Organized,* 1916. *Headquarters,* Philadelphia, Pa. *Meets,* Second Thursday in Month. *President,* H. H. ERICKSON, 1124 Spring Garden St. *Secretary,* H. H. MATHER, 1000 Chestnut St.

PITTSBURGH: *Organized,* 1919. *Headquarters,* Pittsburgh, Pa. *Meets,* Second Monday in Month. *President,* J. F. COLLINS, JR., 435 Sixth Ave. *Secretary,* T. F. ROCKWELL, Carnegie Inst. Tech.

Dues

Members who invite eligible engineers to join the Society should note that: Effective July 1, 1938, dues of new members will be pro-rated on a quarterly basis—for Member and Associate grades $4.50 per quarter, and for Juniors $2.50 per quarter. Initiation fee should accompany all applications.

* * *

Amendments

Amendments to the Regulations Governing the Committee on Research have been mailed to members and are to be voted upon at the Semi-Annual Meeting, June 22, at Hot Springs, Va.

* * *

Ballots

Ballots to vote on the A. S. H. V. E. Standard Code for Testing Stoker Fired Heating Boilers have been sent to members.

* * *

Committee

President Gurney has announced the appointment of an Educational Publicity Committee with the following personnel: J. M. Frank, *Chairman;* C. K. Foster, H. M. Hart, E. P. Heckel, E. N. McDonnell, and J. R. Vernon.

ST. LOUIS: *Organized,* 1918. *Headquarters,* St. Louis, Mo. *Meets,* First Tuesday in Month. *President,* G. W. F. MYERS, 3947 W. Pine Blvd. *Secretary,* D. J. FAGIN, 1344 Woodruff Ave.

SOUTHERN CALIFORNIA: *Organized,* 1930. *Headquarters,* Los Angeles, Calif. *Meets,* Second Tuesday in Month. *President,* E. H. KENDALL, 1978 S. Los Angeles St. *Secretary,* J. F. PARK, 1234 South Grand.

TEXAS: *Organized,* 1936. *Headquarters,* College Station, Texas. *President,* R. F. TAYLOR, 909 Banker's Mortgage Bldg., Houston, Tex. *Secretary,* W. H. BADGETT, Texas Engrg. Experiment Station, College Station, Tex.

WASHINGTON, D. C.: *Organized,* 1935. *Headquarters,* Washington, D. C. *Meets,* Second Wednesday in Month. *President,* L. OURUSOFF, 411 Tenth St., N. W. *Secretary,* L. F. NORDINE, Room 203, 734 Jackson Pl., N. W.

WISCONSIN: *Organized,* 1922. *Headquarters,* Milwaukee, Wis. *Meets,* Third Monday in Month. *President,* J. H. VOLK, 1906 W. St. Paul Ave. *Secretary,* H. C. FRENTZEL, 3000 W. Montana St.

Cooling Requirements for Summer Comfort Air Conditioning in Toronto

By C. Tasker* (*MEMBER*), Toronto, Ont., Can.

A PAPER[1] presented at the 43rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS gave the results of studies made during the summer of 1936 under the Technical Advisory Committee on Comfort Requirements for Summer Cooling at the A.S.H.V.E. Research Laboratory in Pittsburgh, at the Agricultural and Mechanical College of Texas, and at the Ontario Research Foundation, Toronto, Can.

In brief, the findings were that "comfort may be had over a small effective temperature range, which is independent of daily or even weekly variations in outside weather. The results show clearly that with the weather conditions of July, 1936, in Toronto, an effective temperature of from 2 to 3 deg lower was required for comfort than was the case in Pittsburgh. Similar studies made in Texas prove rather conclusively that for their hotter, more continuous and longer summer heat, the same effective temperature is required as was found for Pittsburgh. In Toronto, 70 per cent of the subjects were found to be comfortable over a range of effective temperatures from 68.5 to 71 deg; in Pittsburgh, from 69.5 to 74 deg; and in Texas, from 71 to 74 deg."

The main criticisms levelled against the results of the studies in Toronto were:

(1) They were based on too small a number of observers, and only male observers were used.

(2) In July, 1936, abnormally warm weather persisted over large parts of North America, and in Toronto an all-time high shade temperature was established.

The results obtained in all the studies were criticized on the grounds that:

(3) Higher age groups might react differently from young, healthy men.

(4) Insufficient data were obtained on the reaction to entering the warm outside weather after being in a cooled space.

The 1937 Technical Advisory Committee on Comfort Air Conditioning (consisting of W. L. Fleisher (vice-chairman, Committee on Research), A. E. Beals, F. R. Bichowsky, Thomas Chester, F. E. Giesecke, Elliott Harrington, R. E. Keyes, A. B. Newton, C. P. Yaglou, and C. Tasker, chairman) agreed that the studies should be continued during 1937 to confirm or modify the conclusions drawn from the 1936 and earlier studies. The program which was recommended included:

A. Investigations in the offices of a large building located in Minneapolis, Minn., relative to the comfort requirements of male and female employees of various age groups. (One paper[2] describing these tests was presented at the 44th Annual Meeting of the A.S.H.V.E., and two others are scheduled for presentation at the 1938 Semi-Annual Meeting.)

B. Tests at Texas with particular reference to the transition periods from winter comfort requirements to summer comfort requirements, and vice versa. (The results of these tests are scheduled for presentation at the 1938 Semi-Annual Meeting.)

C. Tests at the Ontario Research Foundation, Toronto, with particular reference to the difference, if any, in the requirements of healthy young men and young women. (This present paper describes the tests made and discusses the results obtained.)

Conditions of Test

As far as possible, the conditions of test were similar to those in the 1936 series.

The subjects reported to the Foundation half an hour before each test commenced, and were sent out into the adjacent park where they stayed until the test was started. During the last 8 or 10 min they walked approximately one-third of a mile at a normal pace. On arrival at the Foundation they recorded their feelings of warmth and degree of perspiration.

Immediately on entering the room, they again recorded their reactions, repeating this at 3-min intervals during the first 15 min, at 5-min intervals during the next 15 min, and at 15-min intervals thereafter until the end of the test.

The *scale of comfort* used in these tests is given and it is similar to that used in the 1936 and earlier tests.

4—Comfortable.
3—Comfortably cool—*i.e.*, the subject was not particularly uncomfortable, but would for the sake of comfort desire a warmer rather than a cooler condition.
2—Cool.
1—Cold.
5—Comfortably warm.
6—Too warm.
7—Hot.

At the same time, the degree of sensible perspiration was observed according to the following scale:

0—Forehead or body, dry.
1—Forehead or body, clammy.
2—Forehead or body, damp (perspiration just visible).
3—Forehead or body, wet (sweat covering the surface, frequently in drops).
4—Perspiration on the forehead runs down, or perspiration on the body runs down or wets through clothing.

Since it was desired to investigate *comfortable condi-*

*Research Fellow, Ontario Research Foundation.

[1]Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker and Carl Gutberlet (A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1936). For presentation at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938.

[2]Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley (A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1937).

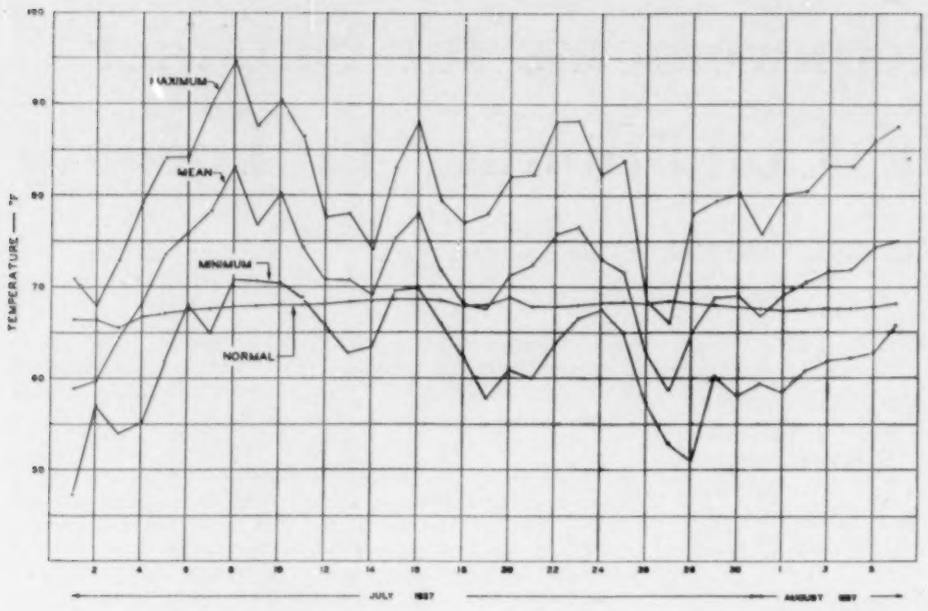


Fig. 1—Log of weather conditions in Toronto, July 1 to August 6, 1937

tions, in none of the tests did the degree of sensible perspiration exceed *2*.

The male subjects occupied themselves by reading, clerical work, and general discussions; the female subjects were more industrious, and sewed or knitted during most of the time, while two of them were engaged intermittently in typing. The men tended to become restless if the test exceeded 1½ hours; the women appeared to be at ease even after 2 hours.

There was no control of the type or amount of food eaten by the subjects either at breakfast or at lunch.

Subjects

Pertinent data on the subjects are given in the accompanying tabulations:

Males

	Age	Weight	Height
A	18 years	141 lb	5 ft 11 in.
B	20 years	170 lb	5 ft 11½ in.
C	18 years	148 lb	5 ft 11 in.
D	22 years	148 lb	5 ft 9 in.

Two of the male subjects were university students; the other two were well-known local amateur athletes.

Females

	Age	Weight	Height
A	26 years	124 lb	5 ft 6½ in.
B	21 years	113 lb	5 ft ½ in.
C	24 years	110 lb	5 ft 1 in.
D	25 years	119 lb	5 ft 5 in.

All four female subjects were university students.

Climatic Conditions

Fig. 1 shows the climatic conditions in Toronto for the period July 1 to August 6; that is, for 2 weeks previous to the commencement, and throughout the period, of the tests. The daily maximum, minimum, and mean temperatures, together with the daily normal temperature (for 70 years) are given. From Fig. 1 it will be seen that from July 5 to July 11 the daily mean temperature was from 6 to 15 deg above the normal, indicating unusually warm weather.

The first six tests with men were conducted on warm days (July 14-16). Six of the other 10 tests with men were made on days when the mean temperature was from 3 to 10 deg below the normal—*i. e.*, cool days.

Of the first 10 tests with female subjects, eight were on days when the mean temperature was from 2 to 8 deg above the normal; all the other eight tests were made on warm days—*i. e.*, 4 to 7 deg above the normal.

Though most of the tests were conducted on warm days, there was nothing to equal the unusually hot weather experienced during the 1936 tests, when for 16 days the daily mean temperature was from 16 to 24 deg above the normal.

Cooling Tests and Heating Tests

Table 1 gives a summary of the tests arranged in chronological order.

Table 1—Summer Comfort Standards
(Summary of Tests Arranged in Chronological Order)

Test No.	Date (1937)	Time of Entry E. D. S. T.	Range of Outdoor Effective Temp. for 2 Hours Prior to Test[a] (Deg)	Outdoor Effective Temp. on Entry[b] (Deg)	Inside Effective Temp. (Deg)	Feeling of Comfort After 30 Min in Test Room
Men						
1	July 14	10:30 a.m.	64–67	67.2	71.5	4.5
2	14	2:00 p.m.	68–72	72.2	72.4	5.0
3	15	9:30 a.m.	70–72	72.3	72.9	4.0
4	15	1:45 p.m.	74–77	77.0	72.5	3.8
5	16	10:00 a.m.	72–76	75.4	72.7	4.5
6	16	2:00 p.m.	77–78	77.7	72.1	4.0
Women						
1	July 19	10:20 a.m.	63–68	66.4	71.7	3.8
2	19	2:00 p.m.	70–72	70.7	71.9	4.0
3	20	9:30 a.m.	64–67	68.3	73.7	4.3
4	20	2:00 p.m.	72–74	72.2	73.2	3.8
5	21	9:30 a.m.	64–69	69.0	71.8	4.0
6	21	2:00 p.m.	75–74	73.4	73.0	4.0
7	22	9:30 a.m.	66–72	71.9	74.2	4.6
8	22	2:00 p.m.	77–78	77.6	73.3	4.5
9	23	9:30 a.m.	69–72	72.8	69.4	4.0
10	23	2:00 p.m	77–78	77.1	70.7	3.7
Men						
7	July 26	9:30 a.m.	60–62	63.8	70.4	4.0
8	26	2:00 p.m.	64–61	62.0	72.4	4.5
9	27	9:30 a.m.	54–56	56.6	68.3	4.0
10	27	2:30 p.m.	59–61	61.2	71.5	4.3
11	28	9:30 a.m.	54–63	63.0	70.0	4.0
12	28	1:15 p.m.	66–69	67.8	72.1	4.1
13	29	9:30 a.m.	61–67	66.8	71.7	4.0
14	29	2:00 p.m.	71–73	72.5	72.7	4.4
15	30	9:30 a.m.	61–66	66.3	72.1	4.0
16	30	2:00 p.m.	72–73	70.5	71.3	4.0
Women						
11	Aug. 3	10:30 a.m.	66–73	72.1	71.7	4.0
12	3	2:00 p.m.	73–76	74.4	72.4	4.1
13	4	9:30 a.m.	66–70	70.3	72.7	4.0
14	4	2:00 p.m.	75–77	75.2	73.6	4.0
15	5	9:30 a.m.	67–71	71.8	73.7	4.0
16	5	2:30 p.m.	76–79	76.9	70.8	4.0
17	6	9:30 a.m.	67–72	73.7	74.0	4.0
18	6	1:45 p.m.	78–79	77.2	74.1	4.4

[a] Based on measurements made at the Dominion Meteorological Station situated approximately ¼ mile from the building where the tests were conducted.
[b] As measured outdoors in the shade, at the entrance to the building, just prior to the subjects entering. No allowance made for wind.

Table 2—Summer Comfort Standards
(Summary of 1937 Tests)

Effective Temp. (Deg)	Feeling of Warmth							
	Indoor Conditions Lower Than Outdoors				Indoor Conditions Higher Than Outdoors			
	Men		Women		Men		Women	
	A*	B**	A*	B**	A*	B**	A*	B**
68								
68½					4			
69				4—				
69½			4			4—		
70				3+	4			
70½			3+		4			
71			4					
71½		4, 4	4	4, 4	4, 4, 4+, 4+	4	4—	
72	4	4		4	4, 4	4	4, 4	
72½	5, 4, 4+	4	4		4+	4+	4	4
73	5—	4+	4	4	4		4—	4, 4
73½			4+, 4				4+, 4	4
74			4+				4, 4+	4
74½		4+						
75								
75½								
76								

*A—30 min. after entering the test room.
**B—45 min after conditions in the test room were changed.

Table 3—Summer Comfort Standards
(1936 and 1937 Tests with Young Men Normally Clothed)

Effective Temp. (Deg)	Feeling of Warmth[a]			
	Indoor Conditions Lower Than Outdoors		Indoor Conditions Higher Than Outdoors	
	1936	1937	1936	1937
66			4	
66½			4—	
67				
67½	3+			
68			3+	
68½	3+			4
69			4	
69½	4—		4, 4, 4	4—
70	4, 4, 4, 4		4+, 5	4
70½			5, 5	4
71	4, 4, 4, 4		4, 4+, 5+	
71½	4	4, 4	6	4, 4, 4, 4+, 4+
72	4, 4, 5, 5, 5	4, 4		4, 4, 4
72½		5, 4, 4+, 4	4+	4+, 4+
73	5, 5, 5, 5, 5	5—, 4+	5, 5+	4
73½			5	
74	5, 5			
74½		4+		

[a] Either 30 min after entering the test room, or 45 min after conditions in the test room were changed.

As in 1936, it was necessary in many of the tests to maintain the test room at a higher effective temperature than the outdoor condition at the time the subjects entered the room. The results have therefore been split up into four groups:

(1) Tests with men—inside conditions cooler than outdoor.
(2) Tests with men—inside conditions warmer than outdoor.
(3) Tests with women—inside conditions cooler than outdoor.
(4) Tests with women—inside conditions warmer than outdoor.

The results of groups (1) and (3) are plotted on Fig. 2, which also shows the results obtained in the 1936 study with men. Fig. 3 shows the results obtained in the 1937 tests in groups (2) and (4), together with the results obtained in similar studies in 1936.

Summary of Reactions

Table 2 gives in summarized form the reactions of the male and female subjects in the 1937 tests to various effective temperatures. The results are divided into two sections.

A. Reactions 30 min after entering the test room from outdoors.

B. Reactions approximately 45 min after the conditions in the room were changed.

In Table 3 the results of the 1936 studies have been added to those of the 1937 studies with young men. From the data given, the following conclusions may be drawn:

(a) When it was *cooler* inside than outdoors,
young men, normally clothed, were comfortable at 70 to 72½ deg ET.
young women, normally clothed, were comfortable at 69½ to 73 deg ET.

(b) When it was *warmer* inside than outdoors,
young men, normally clothed, were comfortable at 69 to 72 or 72½ deg ET.
young women, normally clothed, were comfortable at 69 to 74 deg ET.

Summarizing the 352 votes cast in the 1936 and 1937 studies by both young men and young women, for indoor effective temperatures of 68 to 74 deg regardless of

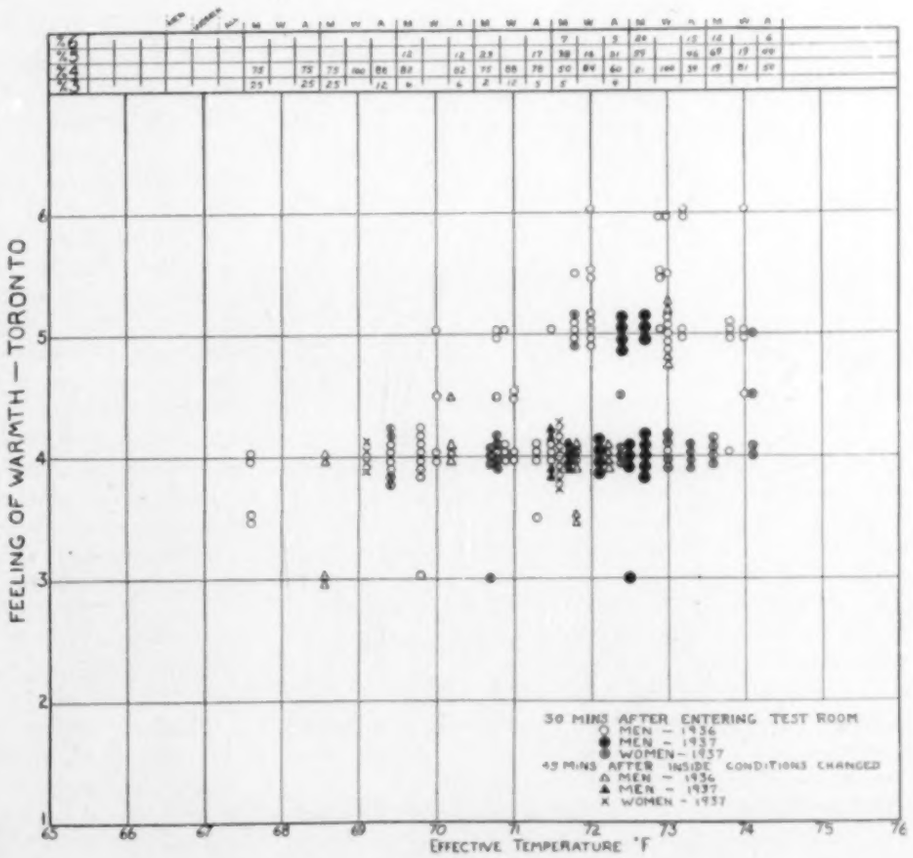


Fig. 2—Relation between effective temperature and feeling of warmth on days when outdoor condition was warmer than test room condition. Percentage of each numerical index of comfort given at top

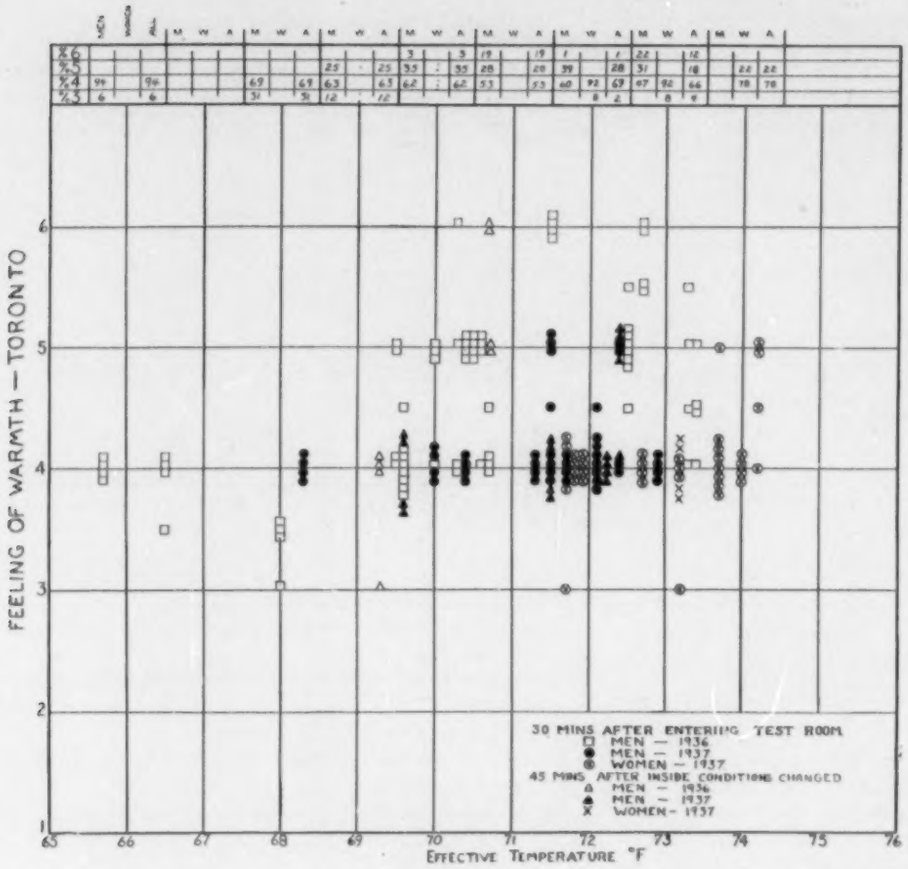


Fig. 3 — Relation between effective temperature and feeling of warmth on days when outdoor condition was cooler than test room condition. Percentage of each numerical index of comfort given at top

whether entered from a warmer or a cooler outdoor condition, the accompanying percentages have been found of *3* (comfortably cool), *4* (ideally comfortable), *5* (comfortably warm).

Indoor Effective Temp.	Percentage of Votes (Warmth Index)[a]		
Deg	*3*	*4*	*5*
68	29	71	..
69	12	80	8
70	2	70	26
71	2	66	23
72	2	64	31
73	3	52	32
74[a]	..	64	33

[a] 75 per cent of the votes cast at 74 deg effective temperature were cast by women subjects.

Comments and Conclusions

From the 1936 tests deductions indicated that young men would be comfortable at 69½-71½ deg ET if they entered from an outdoor condition which was above this range. The 1937 tests seem to indicate that the upper limit of the ideally comfortable zone might be increased slightly to 72 or even 72½ deg ET, but that at 73 deg ET young men, normally clothed, generally felt warmer than desired for comfort.

Young women, on the other hand, were also ideally comfortable at about 70 deg ET, but were comfortable through a wider range, and at 73 deg ET, or even 73½ deg, were not unduly warm.

When the indoor conditions were warmer than outdoor (*i.e.*, at a higher effective temperature), the men in the 1936 tests were comfortable at 69-71 deg ET. The 1937 tests seem to indicate that the upper limit of this range could be extended to 72 deg ET.

For women the limits were not so well defined. Due to lack of time, no tests were made below 71½ deg ET, or above 74 deg. Within this range they seemed to be ideally comfortable with, perhaps, a tendency to desire something a little cooler at 74 deg ET.

Acknowledgment

The author wishes to acknowledge the assistance given by J. Everitt and J. M. McDonell in making these tests and, as chairman of the A.S.H.V.E. 1937 Technical Advisory Committee on Comfort Air Conditioning, to express the Society's thanks to Dr. H. B. Speakman, Director of the Ontario Research Foundation, for making possible both the 1936 series and the present series of tests at no cost to the society.

Test of Washington State Coals with Domestic Stokers

The principal results of 35 burning trials with coals produced in the State of Washington, using overfeed stokers, are given in a report which describes an investigation conducted at the Northwest Experiment Station of the Bureau of Mines, U. S. Department of the Interior, at Seattle, Wash., in cooperation with the University of that State.

The most important observation made during these tests related to a comparison of the over-all efficiencies obtained in continuous and intermittent trials. Intermittent operation, such as is common practice in domestic heating where the stoker operates as demanded by a thermostat, gave the same or only slightly lower overall efficiencies than did continuous operation.

Both caking and noncaking coal in a sized and unsized condition were tested. With noncaking coals the overall efficiencies of the overfeed stoker and hot water boiler installation ranged from 65 per cent at high rates of burning to 75 per cent at low rates. Lower efficiencies were obtained on caking coals, the results indicating that such coals with agglutinating values over 2300 grams were not suitable for use on overfeed stokers.

Until the supply is exhausted, copies of this publication, Report of Investigations 3379, Burning of Various Coals Continuously and Intermittently on a Domestic Overfeed Stoker, may be obtained without charge from the Bureau of Mines, Washington, D. C.

Heat Transfer Through Single and Double Glazing

By M. L. Carr*, R. A. Miller**, Leighton Orr†, (*MEMBERS*), and Alan C. Byers***

IN consideration of the need for data on the relationship of windows and their appurtenances to indoor atmospheric environment which has developed through the increased use of air conditioning, the chairman of the A.S.H.V.E. Committee on Research during 1936 appointed a Technical Advisory Committee on Air Conditioning Requirements of Glass, consisting of M. L. Carr, chairman, R. A. Miller, vice-chairman, F. L. Bishop, A. N. Finn, E. H. Hobbie, R. J. Lillibridge, F. W. Parkinson, W. C. Randall, L. T. Sherwood, J. T. Staples, C. Tasker, G. B. Watkins and F. C. Weinert.

In keeping with a program adopted by this Committee, the Research Laboratory is to make available the results of research carried on by manufacturers or others. This paper represents an analysis and certain deductions made by the Laboratory of data collected, prior to the organization of the Committee, by Pittsburgh Testing Laboratory for the Pittsburgh Plate Glass Co.

It is believed by Col. W. A. Danielson, former chairman of the Committee on Research, that research of a similar character can be carried out under a cooperative plan by the Research Laboratory and research organizations in allied industry, which may greatly expand the useful results of the Society's research.

In accordance with this policy, the Research Laboratory may be called upon to work closely with a manufacturer or anyone else carrying on research, by suggesting a method of attack and procedure for carrying on the work and, if found desirable, to cooperate in the collection and analysis of the data with a view that the Laboratory may use such information in the publication of technical papers.

THE rapid increase in air conditioning applications, together with the demand for high moisture content in the indoor atmosphere, has centered attention on the window as an important source of heat loss by transmission and infiltration. Greater attention has also been directed to such other important effects as condensation and frosting of the inside glass surface.

The transmission of heat through windows forms an important part of the heat loss or gain from or to winter and summer air conditioned space. The values in use by the profession and included for a number of years in THE A.S.H.V.E. GUIDE are probably based upon the results of different investigators. The values contained in the 1937 edition give transmission values of 1.13, 0.45 and 0.28 Btu per square foot per hour per degree temperature difference, for single, double and triple glazed windows, respectively. No differentiation is made in these coefficients depending upon the size of the window panes, the method of installation, size and shape of the muntins separating the individual panes, and other important factors. Neither is there any differentiation in the heat transmission for multiple glazing with different air spaces, with different air velocities, and with varying mean temperature between the inside and outside air. This paper gives information on a number of these points.

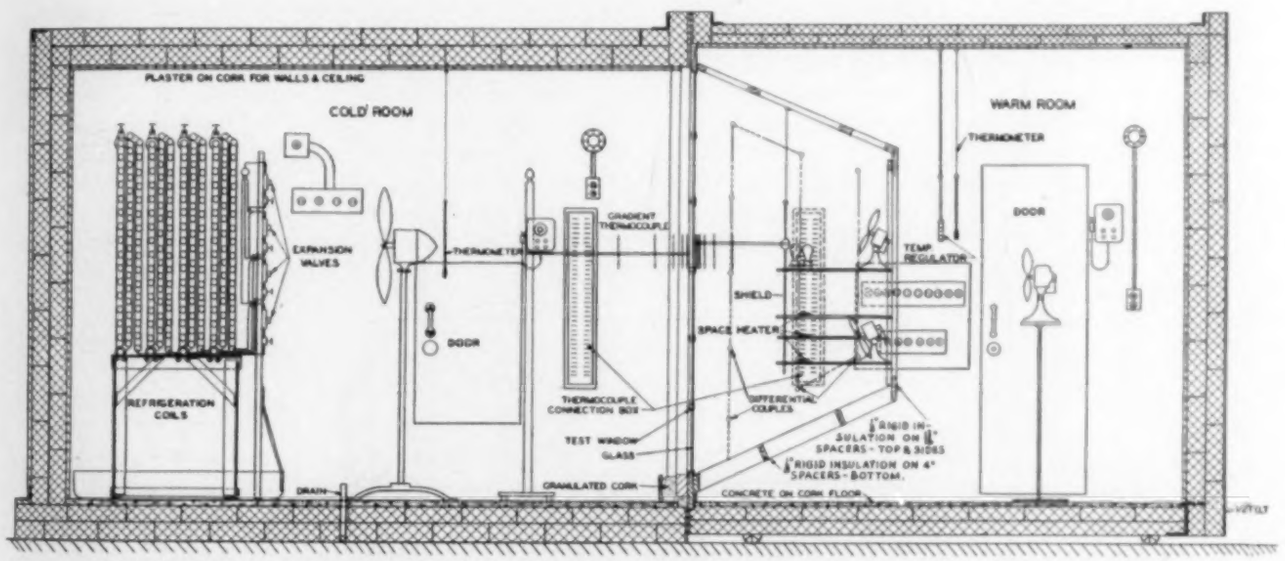


Fig. 1—Vertical cross-section of the test equipment with the 36-light window installed

Apparatus

A cross-section of the heat flow apparatus used in these tests is shown in Fig. 1. It consists of a Cold

*Director, Pittsburgh Testing Laboratory.
**Technical Sales Engr., Pittsburgh Plate Glass Co.
***Research Engr., A.S.H.V.E. Research Laboratory.
†Research Engr., Pittsburgh Plate Glass Co.
For presentation at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938.

Room, in which outdoor air conditions are simulated, and a Warm Room in which indoor air conditions are reproduced, and which contains the guarded enclosure or hot box. The test specimen forms the dividing partition between these two rooms.

The Cold Room is an enclosure about 8 ft by 11 ft by 8 ft high. Its walls, with the exception of the test specimen, and its floor and ceiling are made of 8-in. cork, plastered on the inside. A cork insulated vestibule with refrigerator doors provides access when the open end is closed by the test panel.

The cooling system consists of four independent motor-driven compressor units, each connected to its own cooling coil located in the Cold Room. The capacity of this system is sufficient to maintain a Cold Room temperature of —30 F, with the test panel and a Warm Room temperature of 70 F.

The Warm Room, about 8 ft by 9 ft by 8 ft high, is of similar construction and is mounted on casters to permit its removal from the Cold Room while a test specimen is being installed. A flexible system of electrical heating is provided. It is controlled either automatically with a mercurial thermostat, or manually as desired. Both the Warm Room and the Cold Room are provided with fans for air circulation, electric lights, electric outlets, both 110 and 220 volts, connections for thermocouples and telephones.

The Guarded Enclosure is shaped like the frustum of a four-sided pyramid. Its larger end is open and of the same size as the windows to be tested, and is fastened with a calked air-tight joint to the wood frame in which the test specimen is installed. The walls are of double insulated construction, and contain about 50 differential thermocouples threaded through the walls, with approximately one couple in each 2 sq ft of the inside and outside wall surface. All the differential couples on any one wall of the Guarded Enclosure are in series. Provision is also made to connect all five of these groups in series. The Guarded Enclosure is provided with its own set of heaters, shielded from the test panel, fans and lights. The electric input to all electrical appliances in the Guarded Enclosure is metered.

Temperature measuring apparatus consists of 48 copper-constantan thermocouples, 24 in the Cold Room, 24 in the Warm Room and Guarded Enclosure. Most of the couples are of No. 22 gage B and S, a few are of No. 30 gage. Terminal boxes for the thermocouples are permanently installed in the Warm and Cold Rooms. Cables lead from these boxes to selector switches on the instrument bench. The thermocouple wires and the potentiometer used with them are calibrated against a precision thermometer. The electrical metering apparatus is an integrating watt-hour meter which could be read to one watt-hour. An indicating wattmeter is used as a check. Light threads are used to determine the location and direction of air movement, while a Kata thermometer is used to measure the velocities.

Humidity is measured by a mercurial thermometer type of hygrometer and by a set of wet and dry junction thermocouples placed in the air current from a fan. Some control of the humidity is provided by means of silica gel.

Test Specimens

Two test panels were used in these tests. They were identical in overall dimensions, being made for a window

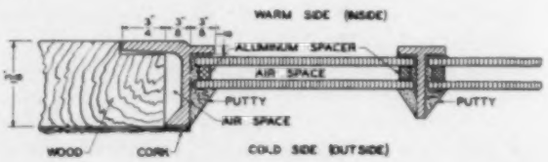


Fig. 2—Detail of test panel—upper side in contact with warm air

opening 7 ft, 4⅜ in. square. They differed only in the size and number of lights. One was for nine lights, $28\frac{13}{16}$-in. square, with $\frac{3}{16}$ to ¼ in. bearing on all sides; the second window had 36 lights, $14\frac{5}{16}$-in. square, with practically the same bearing. Glazing was on the outside. Each sash was installed in the apparatus and then glazed with ⅛-in. (nominal) plate glass, set in ordinary sash

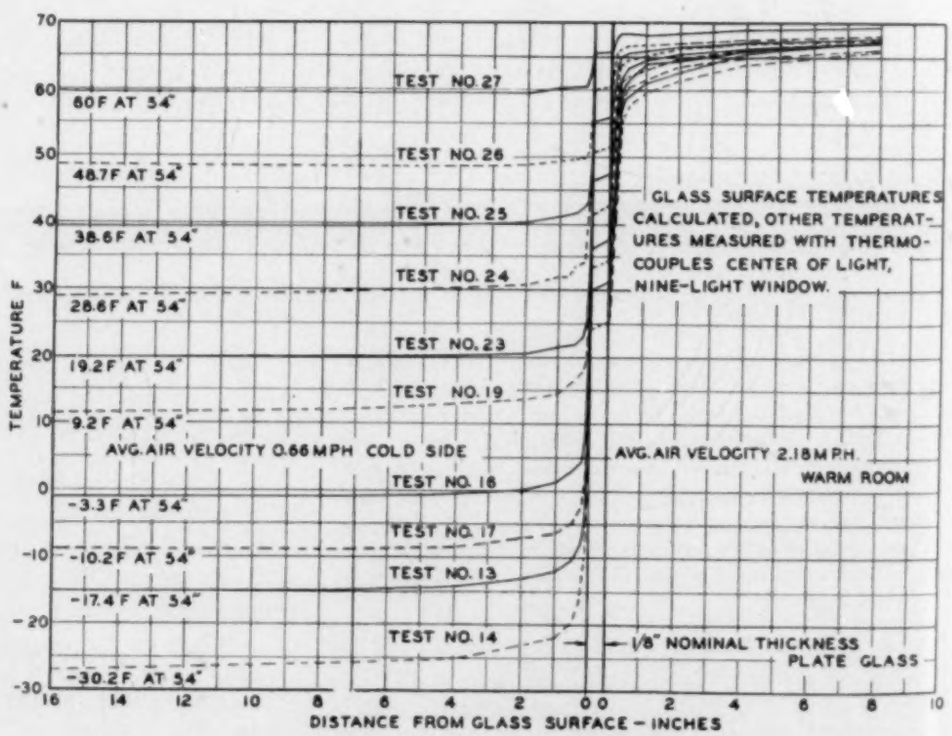


Fig. 3—Temperature gradients through ⅛-in. plate glass single glazed in the 9-light window

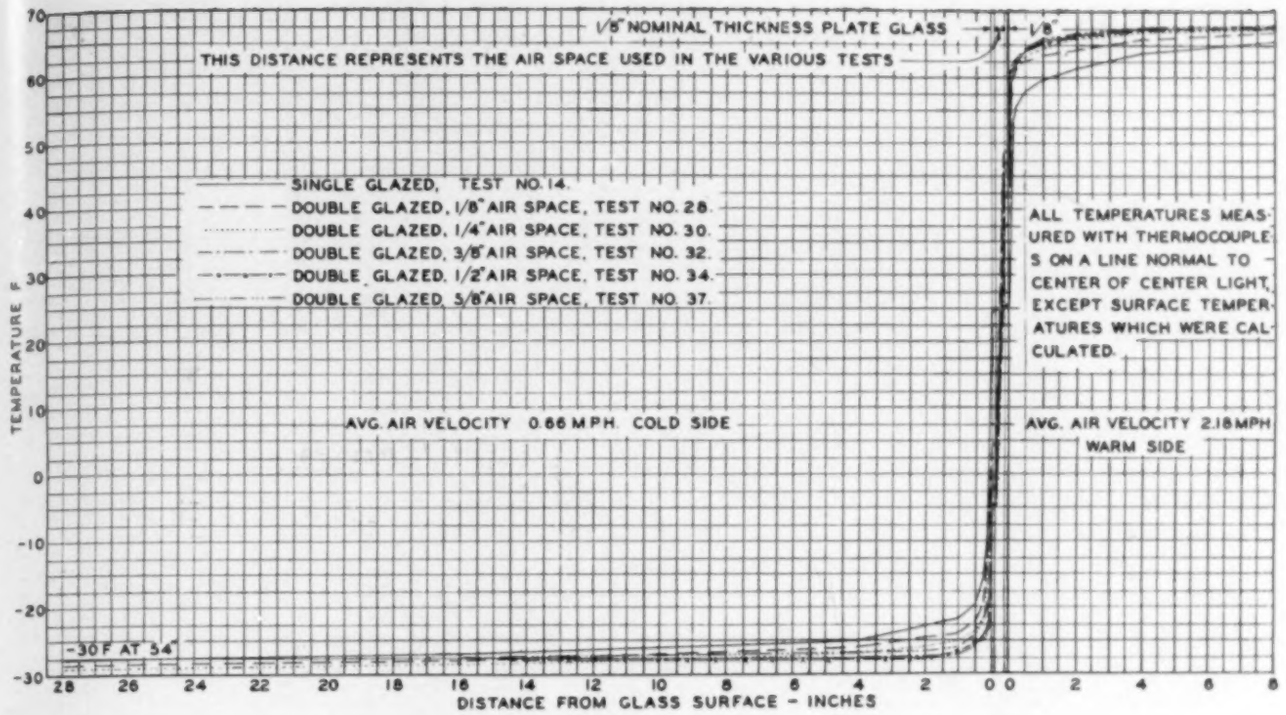

1/8" NOMINAL THICKNESS PLATE GLASS
1/8"
THIS DISTANCE REPRESENTS THE AIR SPACE USED IN THE VARIOUS TESTS
SINGLE GLAZED, TEST NO. 14.
DOUBLE GLAZED, 1/8" AIR SPACE, TEST NO. 28.
DOUBLE GLAZED, 1/4" AIR SPACE, TEST NO. 30.
DOUBLE GLAZED, 3/8" AIR SPACE, TEST NO. 32.
DOUBLE GLAZED, 1/2" AIR SPACE, TEST NO. 34.
DOUBLE GLAZED, 5/8" AIR SPACE, TEST NO. 37.
ALL TEMPERATURES MEASURED WITH THERMOCOUPLES ON A LINE NORMAL TO CENTER OF CENTER LIGHT, EXCEPT SURFACE TEMPERATURES WHICH WERE CALCULATED.
AVG. AIR VELOCITY 0.66 M.P.H. COLD SIDE
AVG. AIR VELOCITY 2.18 MPH WARM SIDE
-30 F AT 54"
TEMPERATURE F
70
60
50
40
30
20
10
0
-10
-20
-30
28 26 24 22 20 18 16 14 12 10 8 6 4 2 0 0 2 4 6 8
DISTANCE FROM GLASS SURFACE - INCHES

Fig. 4—Temperature gradients through 1/8-in. plate glass double glazed with various air spaces in the 9-light window

putty. Fig. 2 shows the window structure tested. Aluminum spacer strips, $\frac{3}{16}$-in. wide, having the thickness of the air space desired, were placed between the lights in the double glazed sash.

Test Procedure

To obtain a desired temperature in the Cold Room, the number of compressors operating and the back pressure of each were adjusted in the light of previous experience. Regulation of the temperature was accomplished by manual manipulation of the back pressure by the regulating valves.

The watt-hour meter was read when thermal equilibrium was established, at half-hour intervals throughout the five-hour duration of the test, and at the end, while temperatures were measured six to eight times during the test. The Warm Room was held at approximately 70 F, and the Cold Room at some definite temperature between —30 F and 60 F. In a series of tests on any sash arrangement the coldest test was run first. No extreme care was taken to attain balanced conditions at some definite temperature, as it was considered more important to adjust the refrigerating equipment to obtain a temperature somewhere near the one desired and then to allow conditions to reach a balance undisturbed by change of regulating valves.

Temperatures were measured at points opposite the center of each light, 1-in. from the glass surface on both the warm and cold sides in the case of the nine-light window. Temperatures for plotting gradients were measured on a line perpendicular to the center of the window, with the thermocouples located in contact with the glass and spaced $\frac{1}{8}$, $\frac{1}{4}$, $\frac{1}{2}$, 1, 2, 4 and 8 in. from the glass on both sides. On the cold side there were also couples at 16 in. and 54 in. from the glass. Two thermocouples were placed in the air space with junctions against the surfaces of the two lights of the double glazed window, on the same perpendicular line with the gradient couples. In case of the wider spacings, $\frac{1}{2}$ and $\frac{5}{8}$ in., one was also placed in the center of the air space.

The same gradient couples were used for the tests with the 36-light window. They could not be located, however, on the perpendicular to the center of the window, since the center was the intersection of two muntins. The light immediately above and to the right of the center was chosen, and the couples were placed on the perpendicular to the center of this light. Also, it was not possible to accommodate sufficient couples to place one opposite each light of the 36-light window. Thirteen were used, disposed as uniformly as possible.

Temperatures were measured at a number of other points in the Cold Room, the Guarded Enclosure, and the Warm Room, at least one mercury thermometer being placed in each. Following each test, all the temperature readings of each thermocouple after equilibrium had been reached were averaged and the averages used in the computations.

Temperature Gradients

A few of the temperature gradients for the tests of the nine-light window are reproduced in Figs. 3, 4 and 5. These are included merely as samples of this type of data, and are characteristic of all the tests. The glass surface temperatures were computed from the air temperature in the rooms, the heat flow, and the film conductance coefficient for the prevailing mean temperature

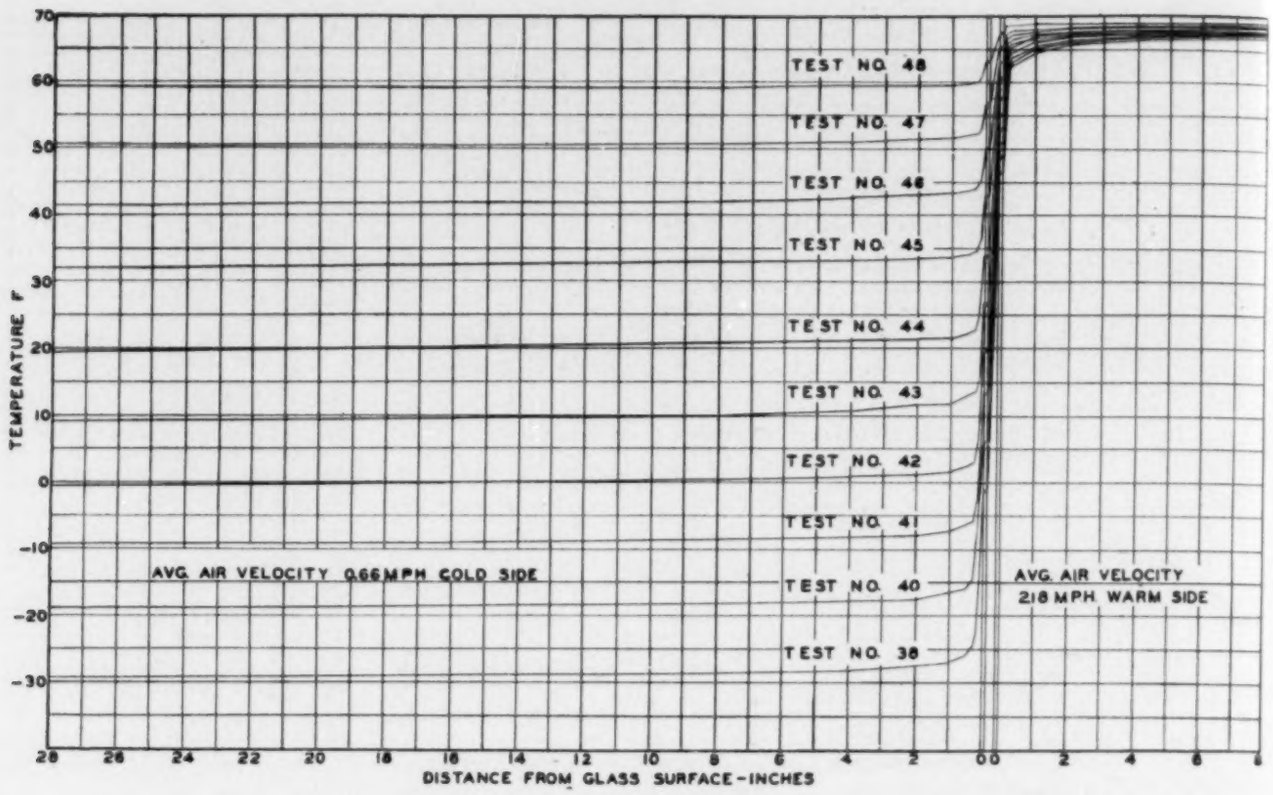


Fig. 5—Temperature gradients through ⅛-in. plate glass double glazed with ¼-in. air space in the 9-light window

as given by the findings of Rowley[1] and the Research Laboratory.[2]

All other points were observed. The largest part of the temperature difference between air and glass in either room occurred within a distance of ¼ in. from the glass surface. The thermal resistance of the glass is small. Comparison of Figs. 3 and 5 shows that inside surface glass temperatures are sensibly higher in the double glazing tests than in the single glazing test. In the single glazed tests these temperatures ranged from 25.5 F to 65.5 F, while in the double glazed tests the range was from 46.5 F to 67.5 F.

Heat Loss

Heat flow data are given in Table 1 and in Figs. 6, 7 and 8. The single glazed, nine-light window was run through a complete series of tests. Tests on the nine-light, double glazed window were run at cold room temperatures of approximately —30 F and 0 F, and with the spacing varied in an endeavor to determine the best spacing, which was concluded to be ¼ in. A complete set was run at this spacing. The 36-light window was tested single glazed with the cold room at approximately —20 F, 10 F and 30 F, and double glazed with ¼-in. spacing at cold room temperatures of —30 F, 5 F, and 30 F.

The transmittance coefficients for the different arrangements of sash, expressed in Btu per hour per degree temperature difference between the air in the warm and cold rooms per square foot of window opening, are given in Fig. 6 for various mean temperatures. These values are plotted against the mean temperature. The curves for the nine-light windows are rather well determined. Those for the 36-light window are less exact, as only three points were determined. These curves were fitted by the method of least distances.

At 20 F mean temperature, the nine-light window, double glazed with ¼-in. spacing, showed only 59.5 per cent of the heat loss of the same window single glazed. At 70 F mean temperature, the loss was 62.8 per cent. For the 36-light window, the figures were 62.6 per cent at 20 F, and 66.1 per cent at 70 F. Curves giving the calculated heat transfer for single and double glass based upon accepted inside and outside conductance coefficients for 2.18 mph and 0.66 mph respectively, and for accepted air space conductances based upon the work of Rowley[3] and the Research Laboratory[4] are also given in Fig. 6.

Fig. 7 shows the variation of heat loss and percentage saving as affected by the thickness of the air space. As would be expected, an increased air space decreased the heat loss. But the curve flattened so rapidly that increases in the width of the air space above ⅜ in. produced relatively small changes in the heat flow. This fact, and the dimensional limits of commercial frames, guided the choice of ¼ in. for the majority of the double glazing tests. It would seem that air spaces between ¼ and ⅜ in. would be most practical. The data for heat flow found in the single glazing tests are plotted for air space equal to zero.

Effects of Air Velocity

All the tests discussed previously were made under the same air velocity conditions; *i.e.*, with the fan in the

[1]Surface Conductances as Affected by Air Velocity, Temperature and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw. (A. S. H. V. E. Transactions, Vol. 36, 1930, p. 429).

[2]Wind Velocity Gradients Near a Surface and Their Effect on Film Conductances, by F. C. Houghten and Paul McDermott. (A. S. H. V. E. Transactions, Vol. 37, 1931, p. 301).

[3]Loc. cit. See Note 1.

[4]Loc. cit. See Note 2.

Table 1—Heat Flow Data

TEST NO.	AIR SPACE IN.	TEMPERATURE, DEG F				DIFF.	ACTUAL MEAN	BTU PER HR	
		WARM ROOM		COLD ROOM				TOTAL	PER DEG F DIFF. PER SQ FT
		ACTUAL	ADJUSTED	ACTUAL	ADJUSTED				
NINE-LIGHT WINDOW SINGLE GLAZED									
14		70.2	70.0	-30.0	-30.2	100.2	20.1	5380	0.990
13		70.4	70.0	-17.0	-17.4	87.4	26.7	4651	0.981
17		70.6	70.0	- 9.6	-10.2	80.2	30.5	4289	0.981
16		72.9	70.0	- 0.4	- 3.3	73.3	36.3	3907	0.983
19		70.9	70.0	10.1	9.2	60.8	40.5	3285	0.996
23		71.3	70.0	20.5	19.2	50.8	45.9	2903	1.053
24		71.1	70.0	29.7	28.6	41.4	50.4	2182	0.974
25		70.7	70.0	38.3	38.6	31.4	55.0	1793	1.053
26		70.6	70.0	49.3	48.7	21.3	60.0	1229	1.084
27		70.7	70.0	60.7	60.0	10.0	65.7	563.5	1.040
NINE-LIGHT WINDOW DOUBLE GLAZED									
28	1/8	70.4	70.0	-29.0	-29.4	99.4	20.7	3517	0.853
29	1/8	70.5	70.0	5.2	4.7	65.3	37.8	2315	0.856
30	1/4	70.8	70.0	-30.2	-31.0	101.0	20.3	3159	0.577
31	1/4	70.2	70.0	1.1	0.9	69.1	35.7	2285	0.610
32	3/8	71.6	70.0	-31.0	-32.6	102.8	20.3	3125	0.562
33	3/8	70.8	70.0	2.3	1.5	68.5	36.5	2059	0.555
34	1/2	69.9	70.0	-30.0	-29.9	99.9	20.0	2966	0.548
35	1/2	70.2	70.0	- 0.9	- 1.1	71.1	34.6	2135	0.553
37	5/8	70.6	70.0	-30.9	-31.5	101.5	19.8	2863	0.520
36	5/8	70.2	70.0	0.1	- 0.1	70.3	35.0	2062	0.542
38	1/4	71.5	70.0	-29.0	-30.5	100.5	21.3	3177	0.583
40	1/4	70.8	70.0	-19.4	-20.2	90.2	25.7	2855	0.584
41	1/4	70.4	70.0	- 9.8	-10.2	80.2	30.3	2626	0.603
42	1/4	71.1	70.0	- 0.2	- 1.3	71.3	35.5	2315	0.599
43	1/4	70.9	70.0	9.3	8.4	61.6	40.1	1998	0.597
44	1/4	70.6	70.0	19.2	18.6	51.4	44.9	1720	0.618
45	1/4	70.8	70.0	32.2	31.4	38.6	51.5	1314	0.627
46	1/4	69.9	70.0	41.4	41.5	28.5	55.7	1005	0.651
47	1/4	69.9	70.0	50.9	51.0	19.0	60.4	678.2	0.658
48	1/4	70.1	70.0	60.2	60.1	9.9	65.1	345.8	0.651
THIRTY-SIX-LIGHT WINDOW SINGLE GLAZED									
52		69.5	70.0	-20.7	-20.2	90.2	24.4	5378	1.099
51		69.6	70.0	9.0	9.4	60.6	39.3	3579	1.089
53		70.0	70.0	34.5	34.5	35.5	52.2	2195	1.141
THIRTY-SIX-LIGHT WINDOW DOUBLE GLAZED									
57	1/4	70.1	70.0	-28.2	-28.3	98.3	21.0	3845	0.884
56	1/4	70.3	70.0	5.3	5.0	65.0	37.8	2475	0.897
58	1/4	70.5	70.0	33.5	33.0	37.0	52.0	1485	0.740

Cold Room blowing at the cooling coils from the center of the room and the two 10-in. oscillating fans in the Guarded Enclosure blowing over the heaters toward the shield. All fans were operated under the identical conditions during heat flow tests as they were when subsequent tests showed average air velocities 1 in. from the lights of 0.66 mph and 2.18 mph in the Cold Room and in the Warm Room, respectively. The heat flow values given in Figs. 6 and 7 are for these velocities.

Early in the course of the tests it became apparent that the velocity of the air passing over the window had a marked influence on heat loss. While not included in the original plan of the investigation, the importance of the subject led to the decision to do some preliminary work with different air velocities. Accordingly, a few temperature measurement tests and heat loss tests were made following the procedure previously described but with different air velocities and of shorter duration. The velocities used in the Cold Room were zero, 0.66 mph and 7.58 mph; in the Guarded Enclosure the velocities were zero and 2.18 mph. These are the average velocities measured 1 in. from the glass with a Kata thermometer. Zero velocity was had with no fans running; actually, there probably was a motion due to convection.

The results of the velocity tests as far as temperature effects are concerned were reasonable and as would be predicted. Air temperature gradients in the Cold Room were flatter with higher air velocities and had a more abrupt transition at the surface film. The distribution of temperatures over the window, measured 1 in. away from the center of each light, was more uniform. In the Guarded Enclosure, the gradients showed no great difference between *still air* and the rather low velocity of 2.18 mph.

The data on heat loss as affected by air velocity are given in Fig. 8. The curves are reproduced from Fig. 6. In all tests represented by the curves, the Cold Room average velocity was 0.66 mph, while the average velocity in the Guarded Enclosure was 2.18 mph. The points for other velocities plotted on the chart represent the heat transfer for the particular window indicated by the symbol and for the air velocities given in the associated parentheses. The velocity in the Guarded Enclosure is given first. The vertical separation between any point and the curve for the same window similarly glazed represents the change in heat flow attributable to the change in air velocities. This does not represent an exhaustive study of the subject, and the data are too

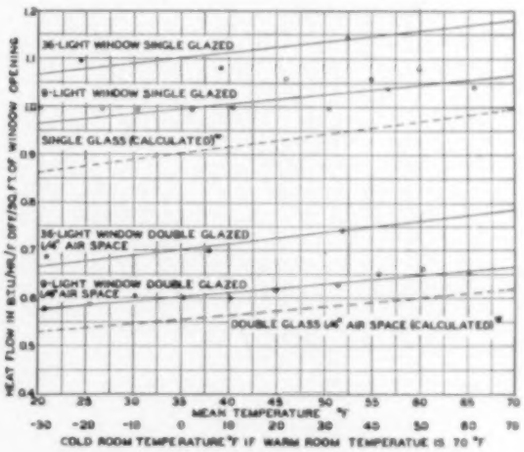

36-LIGHT WINDOW SINGLE GLAZED
9-LIGHT WINDOW SINGLE GLAZED
SINGLE GLASS (CALCULATED)*
36-LIGHT WINDOW DOUBLE GLAZED 1/4" AIR SPACE
9-LIGHT WINDOW DOUBLE GLAZED 1/4" AIR SPACE
DOUBLE GLASS 1/4" AIR SPACE (CALCULATED)*
HEAT FLOW IN BTU/HR./° DIFF/SQ FT OF WINDOW OPENING
MEAN TEMPERATURE °F
COLD ROOM TEMPERATURE °F IF WARM ROOM TEMPERATUE IS 70 °F

Fig. 6—Relation between mean temperature and thermal transmittance of single and double glazed panels

*Calculated curves are based on surface coefficients as given by Rowley[5] and the Research Laboratory[6], and air space conductances given on page 104 of THE A.S.H.V.E. GUIDE 1937.

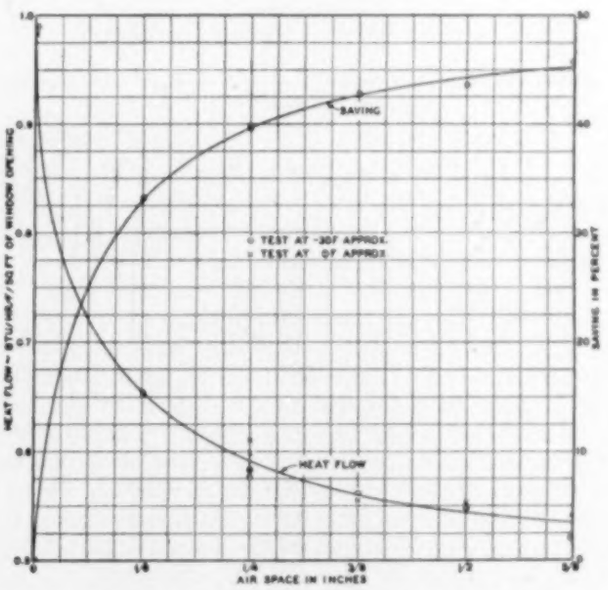


Fig. 7—Thermal transmittance and heat flow saving as functions of air space—9-light window

[5]Loc. cit. See Note 1.
[6]Loc. cit. See Note 2.

scant to permit of any quantitative conclusions. They do indicate the necessity for further work on the subject.

Moisture and Frost Formation

During the tests of the single glazed window considerable difficulty was experienced in keeping frost from forming on the warm side of the window when the Cold Room temperature was —30 F. Silica gel was

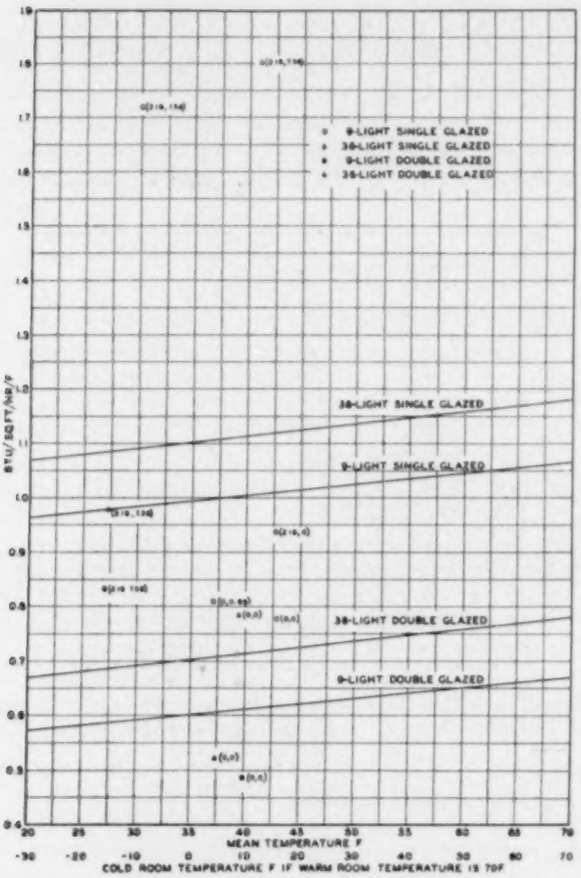


Fig. 8—Effect of air velocity on the thermal transmittance of the windows

used to absorb moisture in the Guarded Enclosure, but it was found that the humidity could not be reduced below 30 per cent. Whenever the temperature of the warm surface of the glass fell below the dew point of the air in the Guarded Enclosure, moisture formed on the glass. This abstraction of moisture from the air lowered the dew point so that only small amounts of dew or frost formed, provided no fresh air was admitted to the Guarded Enclosure by the opening of the door.

Notes made during the single glazed tests show that with a Cold Room temperature of —30 F thin frost was distributed at random over the entire window, the total area covered being about 1 sq ft. At a Cold Room temperature of —20 F, almost all the frost had disappeared. When the Cold Room was at —12 F, and when there was an average velocity of 7.58 mph on the cold side, a very thin film of frost formed on the warm surface over about one-fourth the area of the window. No condensation was noted on the frame. It should be remembered, however, that these observations were made with humidity conditions on the warm side as low as possible, a condition which seldom prevails in buildings.

No frost formed on the warm side of the glass during any of the tests of double glazing, although no precautions were taken to keep the humidity of the Guarded Enclosure low.

Fig. 9 shows the conditions under which moisture or frost will form on single and double glazed windows. The air velocities given are, as previously, in miles per hour, with the values for the Guarded Enclosure given first. These curves do not represent experimentally observed data. They have been derived from the known properties of moist air and are based on the surface temperatures of the tests. These surface temperatures were not measured but computed from the experimentally observed heat flow and other test data.

Some observations were made concerning condensation in the air space of the double glazed windows. At Cold Room temperatures of —30 F, —20 F, —10 F, and —1 F, frost formed on the cold glass surface in the air space of every light. At 9 F the frost was melting, moisture remaining as a thin film. At 19 F all the frost had melted, running drops had formed. At 32 F the moisture occurred as a thin film with a few drops. At 42 F some moisture was still noticeable as a thin film. At 52 F, it was practically all gone, and at 60 F, the window was clear. Twenty-four hours elapsed between the successive temperature changes listed above.

Venting of the air space was accomplished by removing a 1-in. section from the bottom aluminum spacer of each light, the putty being removed to vent the air space to the cold side. One five-hour test at —30 F was run with the vents open. Based on the results of this one test, the heat loss increased between 2 and 3 per cent. The results with respect to the removal of frost from the cold glass surface inside the air space were not as good as expected. Twenty-four hours after the vents had been opened with the Cold Room remaining at —30 F, almost, but not all, of the frost had disappeared from the upper lights, but almost all of it still remained on the bottom row of lights. It was determined later that if the vents were opened before the frost formed results were better. It should also be noted that after resealing the vents almost as much frost formed as previous to opening them, although considerable time was required. The subject of venting should receive further investigation.

Transmittance coefficients for single and double glazed windows resulting from investigations by various authorities vary considerably. Values used in THE A.S.H.V.E. GUIDE since 1922 show variations ranging from 1.03 to 1.13 for single glazed or single sash windows, and from 0.60 to 0.45 for double glazed or double sash windows; the last value in each case being used in THE GUIDE since 1929. John R. Allen[7] gives a value of 0.64; Arthur N. Sheldon[8] gives values of 1.32, 0.88, 0.98, 1.13 and 0.61 for single glazed steel, double glazed steel with ¾-in. air space, double glazed steel with $\frac{3}{16}$-in. air space, single glazed wood, and double glazed wood with $\frac{5}{8}$-in. air space, respectively. Hammond and Holm-

[7]Heat Transmission Through Building Materials, by John R. Allen. (A. S. H. V. E. TRANSACTIONS, Vol. 22, 1916, p. 515).
[8]Heat Transmission Through Various Types of Sash, by Arthur N. Sheldon. (*A. S. M. E. Transactions*, Vol. 38, 1916, p. 869).

berg[9] give transmittance coefficients ranging from 0.97 to 1.16 for variations in mean temperatures from 52 to 86 F. A calculation based upon the film conductance coefficients resulting from work of Rowley[10] and verified by the work of the Research Laboratory,[11] for still air with the prevailing mean temperature and with zero degrees outside and 70 F inside, and accepting what is recognized as a high value for the conductivity of glass, gives 0.688 for a ⅛-in. thickness of glass. The relatively small effect which the resistance of the glass itself offers is indicated by the fact that in completely neglecting the effect of the glass in this computation the coefficient of 0.698 is indicated. Applying a similar computation and using the air space coefficients given in THE GUIDE, two ⅛-in. thicknesses of glass with a ¼-in. air space, gives a coefficient of 0.46.

The considerable variation in these values from different authorities may be accounted for by possible errors in some of the earlier work due to correction for box losses and due to variations in the air velocities actually existing near the glass surface. The calculated values based upon film conductance coefficients are for no air velocity whatever. The values reported by Allen are for velocities existing with no artificial air movement in and around his testing box, while the values reported by Hammond and Holmberg and by Sheldon are for conditions where an unreported artificial air motion was supplied within the box. The values now used in THE GUIDE apply for natural convection within the room and a 15 mph wind outside.

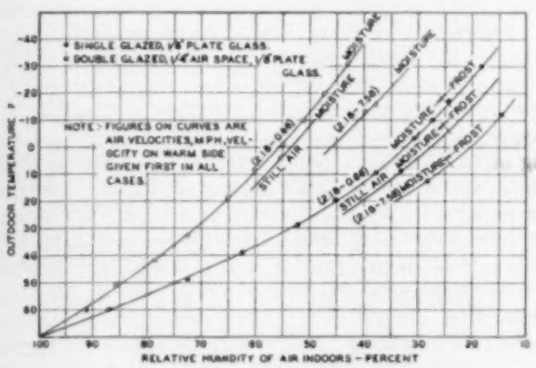

SINGLE GLAZED, ⅛" PLATE GLASS
DOUBLE GLAZED, ¼" AIR SPACE, ⅛" PLATE GLASS
NOTE: FIGURES ON CURVES ARE AIR VELOCITIES, M.P.H., VELOCITY ON WARM SIDE GIVEN FIRST IN ALL CASES.
MOISTURE
FROST
STILL AIR
OUTDOOR TEMPERATURE F
RELATIVE HUMIDITY OF AIR INDOORS - PERCENT

Fig. 9—Curves of moisture formation at various air velocities

The transmittance coefficients for the several window arrangements studied and for several conditions calculated from film conductance coefficients and air space conductances are given for comparison in Table 2. Comparisons of conditions 1 and 2 with 3 for single glazing and 6 and 7 with 8 for double glazing show that calculated values based upon accepted surface coefficients and the same wind velocities on the inside and outside give lower values for glass than found for the panels tested in this study. This would tend to show that, under the conditions of test, the size of the individual lights in the test panel bears a definite relation to the heat transfer, and that the greater the size of the individual lights, the less the loss per square foot.

[9] A Study of Surface Resistance with Glass as the Transmission Medium by H. R. Hammond and C. W. Holmberg. *Journal* of the *A. S. M. E.*, Oct. 1917, p. 849).
[10] Loc. cit. See Note 1.
[11] Loc. cit. See Note 2.

Table 2—Comparison of Results with Those Derived from Calculation

No.	Description	Air Velocity (mph) Warm Side	Air Velocity (mph) Cold Side	Mean Temp. F	Transmittance Btu/Hour/F/Sq Ft	Source of Data
1	36-light single glazed	2 18	0 66	35	1 10	This Paper
2	9-light single glazed	2 18	0 66	35	0 99	This Paper
3	Glass only	2 18	0 66	35	0 90	Calculated[a]
4	Glass only	Still air	15	35	1 14	Calculated[a]
5	Glass only	Still air	15	35	1 13	A.S.H.V.E. Guide
6	36-light double glazed	2 18	0 66	35	0 71	This Paper
7	9-light double glazed	2 18	0 66	35	0 60	This Paper
8	Glass only	2 18	0 66	35	0 56	Calculated[a]
9	Glass only	Still air	15	35	0 63	Calculated[a]
10	Glass only	Still air	15	35	0 45	A.S.H.V.E. Guide

[a] Calculation based on a glass conductivity of 7.2 Btu per hour per inch per degree Fahrenheit per square foot and the film conductance coefficients and air space conductances given by Rowley and the A.S.H.V.E. Research Laboratory as referred to in footnotes 1 and 2.

Conclusions

1. Double glazing reduces heat losses materially as compared with the losses which would take place through the same windows single glazed, under the same conditions of temperature and air movement. The tests made in which only low air velocities were used showed reductions in heat flow ranging from 33 to 46 per cent, depending on the thickness of the air space.

2. Increasing the spacing between the lights up to ⅜ in. showed a rapid decrease in the rate of heat transfer. A further increase in the spacing produced relatively small change in the heat flow. This fact, together with the dimensional limits of commercial frames, indicates that spacing from ¼ in. to ⅜ in. is the optimum spacing for double glazing. The tests showed about a 40 per cent reduction in heat loss with ¼ in. spacing, as compared with the heat transfer for the same window single glazed and under the same conditions of temperature and air movement.

3. The few tests made to show the effect of air velocity indicated a very marked effect in increasing heat transfer through windows. This increase was much greater for the single than for the double glazed windows.

4. Double glazing results in higher indoor glass surface temperatures, as indicated by the absence of moisture or frost formation, thus resulting in greater comfort for the occupants of the room.

L. A. Teasdale on Staff of Yale's Service Bureaus

Announcement has been made recently of the association of L. A. Teasdale as engineer in the University Service Bureaus of Yale University, New Haven, Conn.

Mr. Teasdale, who was formerly with the Office of Hollis French, will devote full time to the University's engineering problems and it is expected that his work will include reports for modernization of old systems, elimination of certain troubles in existing plants, and study of various plans for the more efficient operation of present plants and distribution systems.

Silicosis Hazard Discussed by Dr. Greenburg

Dr. Leonard Greenburg, of New York, addressing a session of the *American Foundrymen's Association* at Cleveland, O., May 19, stated that the silicosis hazard in the foundry industry in New York State is of mild severity and may be expected to yield to appropriate measures of control. The incidence of this disease, as disclosed by a study of more than 4000 foundries in New York State indicated that approximately 2.7 per cent of the 12,000 foundry workers in the state are affected with silicosis. Of these, 2.2 per cent were in the first stage; 0.4 per cent were in the second stage; and 0.1 per cent were in the third stage silicosis. The differentiation of cases of silicosis in the first, second and third stages is arbitrary, but is useful as an indication of the extent to which the condition has advanced.

CORRELATING THERMAL RESEARCH

As a part of the efforts of the A.S.H.V.E. Research Laboratory to correlate research in thermal engineering carried on by the many institutions engaged in such work, and to disseminate the published results of such studies together with other reports of progress in the field, and in order to make this information available to the membership of the Society, there is published monthly on this page a limited number of brief abstracts of articles which it is believed will be of interest to all concerned. For more complete lists address the Librarian, A.S.H.V.E. Research Laboratory, U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.

F. C. Houghten, *Director*

W. L. Fleisher, *Chairman*
COMMITTEE ON RESEARCH

• Deficiency of Atmospheric Humidity as Contributing Factor in Prolonged Asthma, by S. J. Parlato. *Journal Allergy,* Vol. 8, pp. 566-572, Sept., 1937. Report of tests made during cold months. A low relative humidity in warm rooms is an important factor in causing asthmatic attacks.

• Ceiling Heating Systems. *Sulzer Technical Review,* No. 4, 1937. Panel warming in the ceiling considered more comfortable than floor warming. Also the system may be used with cold water for a summer cooling system.

• An Electrostatic Dust Weight Sampler, by E. C. Barnes and G. W. Penney. *Journal of Industrial Hygiene and Toxicology,* Vol. 20, No. 3, March 1938, pp. 259-265. Description of an electrostatic dust sampler used by the authors for sampling various dusts and fumes. The sampler consists of a small blower to draw the sample of air, the precipitation chamber and a power supply which furnishes a d-c voltage, all mounted in a case. The complete instrument weighs 37 lb.

• Ventilating the Lincoln Tunnel, by C. F. Greeves-Carpenter. *Refrigerating Engineering,* Vol. 35, No. 2, Feb. 1938, pp. 87-91. Description of new Lincoln Tunnel under the Hudson River at West 39th Street, New York City. South tube is open to traffic. Some details of construction and of the transverse ventilating system. Fresh air is blown into floor level of the tunnel and exhausted from the ceiling grilles in order to avoid horizontal flow of air in the tube.

• Some Features of the Ventilation of the Mersey Tunnel, by J. R. D. Walker. *The Colliery Guardian,* Vol. CLVI, March 11, 1938, pp. 443-444. Description of method of air flow into and out of the tunnel; also description of ventilating plant which has a total capacity of over 10 million cfm. CO analyzer and visibility indicators are provided. It was found that before CO became objectionable the atmosphere was seriously polluted by oil and soot from exhausts. Also comparison of some of the characteristics of the Mersey, Holland and Antwerp Tunnels.

• Elimination of Algae in Cooling Towers. *Heating and Ventilating,* Vol. 35, No. 3, March 1938, pp. 51-52. A description of method of treating recirculated water in atmospheric cooling devices with chlorine gas. Treatment consists of injecting chlorine gas into the water, strong enough to kill the algae, once or twice a week. Gas dissipates to atmosphere in a few hours and thus does not corrode the pipe.

• Remarks on Air Conditioning for Mines, by J. F. Kooistra. *Mining Congress Journal,* Vol. 23, No. 11, Nov. 1937, pp. 23-25. General discussion of air conditioning in mines. Discussion of effective temperature conditions in mines and how improvement may be obtained.

• Dew-point Potentiometer for Determining the Moisture Content of Gases, S. S. Stack. *General Electric Review,* Vol. 41, No. 2, pp. 106-108; 1938. A new portable instrument for making measurements of the amount of moisture in furnace atmosphere. The dew-point temperature of a small metallic mirror in contact with the gas, is converted into moisture content measurement by means of a chart.

• A Differential Hot Wire Gaseous Flow Meter, Marcel J. E. Golay. *The Review of Scientific Instruments,* Vol. 9, No. 2, Feb. 1938, pp. 55-57. Description and mathematical derivation of equations representing the conditions in a differential hot wire gaseous flow meter which is essentially linear in response with indications reversing when the direction of gaseous flow is reversed. It has all the desirable characteristics of the hot wire instrument of conventional design.

• The Use of Pipe Bends as Flow Meters, by Herbert Addison. *Engineering,* Vol. CXLV, No. 3764, March 1938, pp. 227-229. Report of tests conducted by the author on the use of bends as flow meters. Mathematical discussion of data obtained where water is used as the liquid. Data were taken on rectangular, circular and square-section bends. Bends as flow meters involve no additions or important alterations to an existing pipe system, and therefore no additional loss of head. They should not be used where velocity in pipe is less than 4 fps; the differential head becomes too small for reliable measurement.

• New Insulation Studies, by E. R. Queer and F. G. Heckler. *Refrigerating Engineering,* Vol. 35, No. 3, March 1938, pp. 167-170 (to be continued). Summary of results obtained and test methods used in study of insulating material for low temperatures. Thermal conductivity tests made by use of the guarded hot plate method. Other properties of insulating material of interest are: compressive strength and recovery, transverse strength, ease of handling, resistance to impact, dimensional uniformity, moisture absorption and capillarity, fire resistance and settling of fill materials. Test methods to determine these qualities are given for 16 materials, including cork board, fiber boards and glass wool.

• Tentative Code of Minimum Requirements for Comfort Air Conditioning, by Joint Committee of *A.S.R.E.* and A.S.H.V.E. *Refrigerating Engineering,* March 1938, Vol. 35, No. 3, pp. 186-188.

• Powdered Fuel Firing. A Means of Increasing the Output of Lancashire Boilers, by T. F. Hurley and R. Cook. *The Institute of Fuel,* Vol. XI, No. 57, Feb. 1938, pp. 195-208. Report of tests showing the possibility of obtaining high outputs with a Lancashire boiler. Basic principles and construction of some of the apparatus used in the tests. Detail of burners used and of the pulverized fuel distributor.

• Symposium on Fuels for Heating and Hot Water Supply. *The Institute of Fuel,* Vol. XI, No. 57, Feb. 1938, pp. 167-194. Fuel Oil for Heating and Hot Water Supply, by I. Lubbock. Other Fuels for Heating and Hot-Water Supply, by H. L. Pirie. Description and diagrams of layout for heating systems for various types of buildings, using oil, various kinds of coal and coke for fuel. Paper deals with hand fired and stoker furnaces.

• Scientific Research and Smoke Abatement, by Prof. R. Whytlaw Gray. *The Journal of the National Smoke Abatement Society.* Vol. 9, No. 33, Feb. 1938, pp. 11-13. Discussion of research on Physical Aspects of Smoke Pollution at the University of Leeds Chemical Department.

• Tinted Windows to Kill Sun Glare. *Chemical Industries,* Vol. 39, No. 2, p. 149, 1936. Opaltone for tinting window glass to kill glare and still admit more than 90 per cent of the daylight. Room temperatures are reduced as much as 10 to 15 deg through the use of opaltone. It is available in two forms, either permanent or removable, blue or white.

• Information on Ultraviolet Transparency of Window Materials and Fabrics. *U. S. Department of Commerce, National Bureau of Standards,* Letter Circular LC-511. Jan. 3, 1938, 2 pages. References to five papers on ultraviolet transparency of window materials, prepared by U. S. Bureau of Standards. Also some data on the percentage transmission of various window glasses when new and after exposure to a quartz lamp and to the sun.

• Regulation of Heat Loss from the Human Body, by James D. Hardy and Eugene F. Dubois. *Proceedings of the National Academy of Science,* Harvard School of Public Health, Boston,

Mass., Dec. 1937, Vol. 23, No. 12, pp. 624-631. Report of studies based on the heat loss of two normal nude men in a large respiration calorimeter. Temperature range from 73.4 to 95 F. Results show rates of heat loss by radiation, convection and vaporization. Also the relation of skin and rectal temperature with increase of air temperatures. The chill produced by lowering body temperature is noted.

• Detroit's Climate—Its Relation to Seasonal Cooling Load, by L. W. Jillson and P. S. Hosman. *Heating and Ventilating*, Vol. 35, No. 4, April 1938, pp. 31-36. Discussion of data taken from the dry- and wet-bulb readings available from the hourly official weather records in Detroit, Mich. Extreme conditions which govern the capacity of the cooling system and average conditions which have the most influence on the seasonal cost of cooling are given. Tables also give hours of sunshine and velocity of average and extreme winds in Detroit.

• Effects of Ultraviolet Radiation on Air-Borne Bacteria. *Heating and Ventilating*, Vol. 35, No. 4, April 1938, p. 37. Description of use of ultraviolet radiation lamp developed by H. C. Rentschler and R. F. James in killing air borne bacteria. This lamp is effective in air conditioning air ducts; also in sterilizing hospital operating rooms.

• New Moisture Meter. *The Parliament of the Commonwealth of Australia*, Eleventh Annual Report of the Council for Scientific and Industrial Research for the year ended June 30, 1937, No. 89—F. 5182. Report of the development of an electric moisture meter. It is used in lumber mills to determine the moisture of boards coming from the planing mill. Full description of meter in Aug. 1937 issue of the Council's *Journal*.

• Modernizing for Comfort in a 200 Year Old Farmhouse. *Plumbing and Heating Journal*, Vol. 104, No. 4, April 1938, pp. 30-34. Description of a modern air conditioning system installed in a 200 year old farmhouse near Westport, Conn. Oil fired radiator heat and modern plumbing make old house comfortable.

• Temperatures in Drying—Their Effect on Cost of Operation and Wear Resistance of Textiles, by B. R. Andrews. *Mechanical Engineering*, March 1938, pp. 231-234. Report of effect of various drying temperatures, during manufacturing processes, upon durability and quality of finished textiles.

• Moisture Difficulties in Modern Construction, by E. R. Queer, *Heating and Ventilating*, Vol. 35, No. 3, March 1938, pp. 63-65. Discussion of the economic value as well as the comfort obtained from heat insulation and air conditioning. Value of winter double windows. Table of temperature within walls with various types of insulation. Methods of preventing condensation on windows and inside walls.

• Conditioning Feedwater for Steam Generators, by Charles E. Joos. *Mechanical Engineering*, Vol. 60, No. 3, March 1938, pp. 223-230. Description of methods of conditioning boiler feedwater and equipment used to meet the requirements of high pressure boilers operated at high ratings, from the plant operator's point of view.

• The Lincoln Vehicular Tunnel, New York. *Engineering*, London, Vol. CXLV, No. 3769, April 8, 1938, pp. 375-377. Discussion of traffic conditions in New York City before and after the construction of the tunnels under the Hudson and the Washington Bridge over the Hudson. Description of the Lincoln Tunnel, construction methods and design of air conditioning layout showing fresh air ducts under the roadway and exhaust air ducts over the ceiling of roadway.

• Heliostat Illumination of Buildings. (Windowless). *The Engineer*, London, Vol. CLXV, April 15, 1938, pp. 426-427. Description of a method of using sunlight to light a complete building without the use of windows. A system of mirrors reflects the sunlight down a tunnel in center of the building and other mirrors reflect the light to ceiling mirrors in the individual rooms. Electric lights are automatically turned on when sunlight is not strong enough to light the building.

• Air Conditioning Factors, by Thomas Chester. *Journal of the Institution of Heating and Ventilating Engineers*, Vol 5, No. 60, Feb. 1938, pp. 538-592. Discussion of the basic facts involved in the production and automatic maintenance of suitable atmospheric temperature, relative humidity, freshness, air motion at a desirable rate and elimination of air-borne dust and dirt.

• Air Conditions for Growing Mushrooms. *Heating and Ventilating*, Vol. 35, No. 3, March 1938, p. 52. Discussion of mushroom growth, atmospheric conditions desirable for growth of the mycelium or spawn and the production of commercial mushrooms.

• Summer and Winter Air Conditioning of Automobiles and Buses, by L. W. Child, *Heating and Ventilating*, Vol. 35, No. 3, March 1938, pp. 38-40. Discussion of difficulties experienced in properly air conditioning automobiles and buses due to small space allotment per person, and to small space for equipment and variable speed of vehicle producing various air leakage conditions.

• Air Conditioning—Its Recent Trends and Developments, by Howard E. Degler. *Ice and Refrigeration*, Vol. 94, No. 4, April 1938, pp. 279-282. Discussion of general trends in air conditioning and the presentation of information useful to all persons interested in the design, manufacture, marketing, installing, servicing and use of air conditioning equipment.

• Coal, Coke, Anthracite and Graphite Research. *Colliery Guardian*, Vol. 15, No. 169, March 1938, pp. 87-90. Discussion of some researches under way in various laboratories, on the nature, properties and the constitution of coal.

• Effect of Yaw on Vane Anemometers, by Roy H. Heald and Paul S. Ballif. *U. S. Bureau of Standards*, Research Paper *RP* 1056, 8 pages. A yaw of 15 per cent (the angle between the axis of the instrument and the direction of air flow) produced an error in anemometer readings of from 1 to 5.5 per cent depending on the design of the anemometer.

• Notes on Air Regulators, by Fred Lebeter. *Colliery Engineering*, Vol. 15, No. 170, April 1938, pp. 112-132. Discussion of use of air regulators to apportion the air in various parts of a coal mine. Relation of regulator air resistance to the resistance of a certain length of roadway. Mathematical derivation of formula for this relationship.

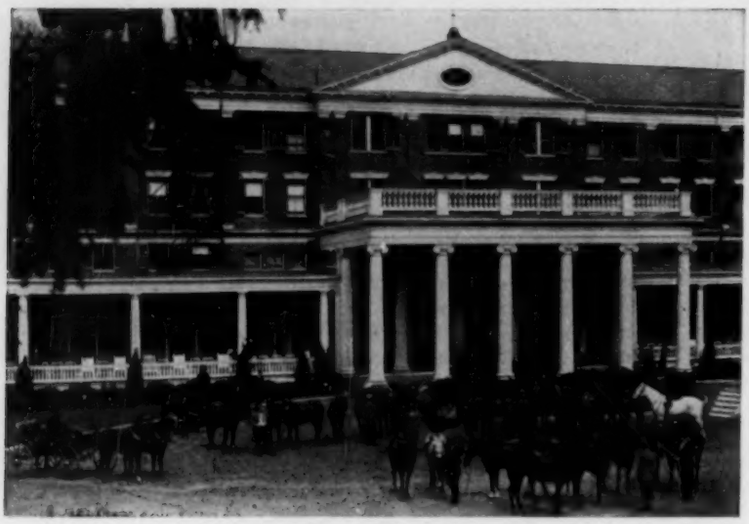
• Heat Emission from Radiators, by K. F. Rubert. *Cornell University Engineering Experiment Station, Bulletin* 24, Ithaca, N. Y., 53 pages. Report of a study showing that under ordinary conditions 40-50 per cent of heat emission is given off by radiation from the radiator commonly used to heat buildings.

• Oil Burning, by H. A. Romp. Published by Martinus, The Hague. 336 pages. The historical development of oil burning in Russia and in America, also early development in western Europe. Recent development of domestic oil burning. Basic principles of oil burning. Physical and chemical properties of fuel oil. Modern forms of oil burning devices.

• A Few Facts—A Few Figures about 1937. Committee of Ten of Coal and Heat Industries, Information Bulletin, Mar. 31, 1938. Summary and review of happenings in the coal and allied heating equipment industries in 1937 reveal substantial progress. Data on fuel production including natural and manufactured gas, also sales and retail value of electrical merchandise.

• Reference Tables for Thermocouples, by William E. Roeser and Andrew I. Dahl. *Journal of Research, U. S. Bureau of Standards, R. P.* 1080, March 1938. Tables have been prepared, after making measurements on representative samples of iron, constantan and copper, to show properties affecting temperature readings on thermocouples made from these wires. If the new tables are accepted by users and manufacturers of thermocouples, the problem of replacing thermocouples will be greatly simplified.

• Factors in Estimating Summer Air Conditioning. *Automatic Heat and Air Conditioning*, Vol. 9, No. 4, April 1938, pp. 32-35 and 74-77. Description of an office air conditioning system. An analysis of the installation of the system, the nature of the work encountered, the manner in which the air conditioning values were figured and the method of actual installation. Detailed computation of heat gain through the walls, sun effect and electrical heat produced in the room, also sensible heat produced by occupants. Determination of refrigeration required.



On to Hot Springs for June Meeting

A THREE-DAY meeting of the Society will be held June 20-22 at The Homestead, Hot Springs, Va., and members will have an opportunity to enjoy the scenic beauties of the Alleghenies and visit the historic spots of Virginia, as well as participate in the technical sessions and entertainment program planned by the Committee on Arrangements.

The program provides for Council and committee meetings, in advance of the technical sessions, so that many of the officers and members will arrive in Hot Springs on the 18th and 19th of June. Elaborate plans are being made for the entertainment of ladies and a children's program is planned for the younger members of the family.

Service at The Homestead is on the American plan and the daily rate includes room with bath and three meals. There will be no extra charge for the banquet to guests of The Homestead. Members should make early reservations on the special cards provided. In double rooms the rate is $9.00 per person per day; single rooms $10.00 per person per day. Adjusted rates for children under 15 years of age are available on application.

For those traveling by train convenient schedules and through car service are available from many Chapter cities. Special cars will be provided by the C. & O. Railroad and connecting lines whenever a group of 14 or more members desire to form a party.

For those who wish to motor there are excellent roads to Hot Springs from all parts of the country. Typical routes from several cities are:

From Toronto and Buffalo:

Route 2—Burlington, Hamilton.
Route 8—St. Catharines, Niagara Falls.
U. S. 62—Buffalo, Hamburg.
U. S. 219—Salamanca, Bradford, Kane, Johnsonburg, Ridgway, Brockway, Du Bois, Grampian, Burnside, Barnesboro, Carrolltown, Ebensburg, Johnstown, Ferndale, Jennerstown, Somerset, Berlin, Meyersdale, Salisbury.
U. S. 40—Frostburg, Cumberland.
U. S. 220—Keyser, Junction, Petersburg, Monterey, Warm Springs, Hot Springs.

From Philadelphia:

U. S. 30—Lancaster, York, Gettysburg.
Penn. State 116—Monterey.
Md. State 60—Hagerstown.
and
U. S. 40—Cumberland.
U. S. 220—Keyser, Junction, Petersburg, Monterey, Warm Springs, Hot Springs.
or (via Skyline Drive)
U. S. 11—Martinsburg, Winchester, Harrisonburg, Staunton.
U. S. 250—Churchville, Monterey.
U. S. 220—Warm Springs, Hot Springs.

From Cincinnati and Charleston:

U. S. 52—Manchester, Portsmouth, Huntington.
U. S. 60—St. Albans, Charleston, Ansted, Lewisburg, Covington.
U. S. 220—Hot Springs.

Plans should be made now to attend the Semi-Annual Meeting, June 20-22.

Method of Choosing Location of, Financing and Conducting Meetings of the Society

Resolved: That inasmuch as the Annual and Semi-Annual Meetings of the Society come under the jurisdiction of the Council, the following rules governing the handling of such meetings be adopted by the Council and published in the JOURNAL of the Society at least twice during every year, preferably just prior to each meeting:

1—The Council will select the city in which the Annual or Semi-Annual Meeting is to be held, giving due consideration to the invitations received from Chapters or members as well as to the advisability of so distributing those meetings as to make them of the greatest advantage to the general membership, and to reduce as far as possible the expense of members attending.

2—That an appropriation be made to cover the entertainment or local expenses, incurred in connection with the meeting not exceeding $500.00, the regular meeting expense to be taken care of by the General Fund of the Society in the regular way.

3—That no registration fee or compulsory obligations of any nature be imposed on members or guests.

4—That the purchase of tickets for banquets or for any other form of entertainment that may be provided be entirely voluntary.

5—That the grouping of features and the sale of tickets for group features be discouraged.

6—That the raising of funds from manufacturers of heating apparatus be discouraged.

7—That the display of samples, or of literature, advertising the product of any manufacturer in any way, shape or form, be not permitted at the booths, registration desk, or in or about the meetings.

8—That the distribution of trade papers be entirely at the discretion of the committee in charge.

9—That the local Chapter or local members, be empowered to form a General Committee with such sub-committees as may be required to handle the details of transportation, hotel accommodations, entertainment, finance, etc., and that this General Committee be requested to confer frequently with the Council, through the Secretary of the Society, and to make frequent reports on progress in connection with the various matters being handled by them.

10—That the arrangements of elaborate and costly entertainment features be discouraged.

Adopted at Council Meeting, January 29, 1926.

PROGRAM

Semi-Annual Meeting, June, 1938

(All Events Eastern Standard Time)

Sunday, June 19

10:00 a. m. Meeting of Committee on Research.
2:00 p. m. Meeting of the Council.
2:30 p. m. Committee on Air Conditioning Requirements of Glass.

Monday, June 20

8:30 a. m. Registration.
9:30 a. m. Technical Session:

1. A Test Method for Air Cleaning, by R. S. Dill.
2. The Flow of Air Through Exhaust Grilles, by A. M. Greene, Jr. and M. H. Dean.
3. The Condensation Nuclei Content of the Air as Related to Air Freshness, by R. A. Nielsen.
4. Study of Summer Cooling in the Research Residence Using a Small Capacity Mechanical Condensing Unit, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick.

2:00 p. m. Committee on Air Distribution and Air Friction.
2:00 p. m. Committee on Radiation and Comfort Winter and Summer.
2:00 p. m. Committee on Weather Design Conditions.
8:00 p. m. Committee on Summer Air Conditioning for Residences.
8:00 p. m. Guide Publication Committee Meeting.

Tuesday, June 21

9:30 a. m. Technical Session:

1. Heat Transfer Through Single and Double Glazing, by M. L. Carr, R. A. Miller, Leighton Orr and Alan C. Byers.
2. Radiation and Convection from Surfaces in Various Positions, by G. B. Wilkes and C. M. F. Peterson.
3. Performance of Surface Coil Dehumidifiers for Comfort Air Conditioning, by G. L. Tuve and L. T. Seigel.
4. The Measurement of Heat Transfer by Free Convection from Cylindrical Bodies by the Schlieren Method, by L. M. K. Boelter.

2:00 p. m. Committee on Insulation.

Wednesday, June 22

9:30 a. m. Technical Session:

1. Cooling Requirements for Summer Comfort Air Conditioning in Toronto, by C. Tasker.
2. Seasonal Variations in Effective Temperature Requirements, by F. E. Giesecke and W. H. Badgett.
3. Shock Experiences of 275 Workers Upon Entering and Leaving Summer Cooled and Conditioned Offices, by A. B. Newton, F. C. Houghten, Carl Gutberlet, R. W. Qualley and M. C. W. Tomlinson.
4. The General Reactions of 275 Office Workers to Summer Cooling and Air Conditioning, by F. C. Houghten, A. B. Newton, R. W. Qualley and Edward Witkowski.

2:00 p. m. Committee on Heat Requirements of Buildings.

Entertainment

Monday, June 20

11:30 a. m. Concert for Ladies.
2:00 p. m. Golf Tournament (Cascade Course) (18-Hole Flag Tournament).
2:30 p. m. Tennis, Croquet, Horseshoe Pitching. (Sign up at Registration Desk.)
2:30 p. m. Carriage Ride.
7:00 p. m. Get-Together Dinner.
8:00 p. m. Movies in Auditorium.
8:30 p. m. Bridge Party.
10:00 p. m. Dancing (Crystal Room).

Tuesday, June 21

10:30 a. m. Ladies Golf and Putting Contest.
2:00 p. m. Golf Tournament (Homestead Course) (18-Hole Medal Play) (Research and Chapter Cups).
2:00 p. m. Motor Trip and Inspection of Covington Paper Mill.
2:30 p. m. Ladies Bridge and Tea.
7:00 p. m. Semi-Annual Banquet and Dance.

Wednesday, June 22

2:00 p. m. Golf Tournament (18-Hole Kickers Handicap).

Committee on Arrangements

J. A. Donnelly, *General Chairman.*
E. W. Klein, *Finance Committee.*
J. K. Peebles, Jr., *Publicity Committee.*
J. R. Shanklin, *Banquet Committee.*
L. F. Nordine, *Golf Committee.*
I. B. Helburn, *Registration Committee.*
R. McC. Johnston, *Entertainment and Sports Committee.*

Chicago Supplement for 1938
to
Code of Minimum Requirements for Comfort Air Conditioning

Supplement prepared by representatives of the following organizations: American Society of Heating and Ventilating Engineers (*Illinois Chapter*); *The American Society of Refrigerating Engineers (Chicago Section); Chicago Association of Consulting Engineers; Chicago Master Steamfitters' Association; Illinois Society of Architects; The Refrigerating Machinery Association (Chicago Section); Ventilating and Air Conditioning Contractors' Association of Chicago.*

Sections and paragraphs refer to corresponding sections and paragraphs in the Code.

SECTION II—Paragraph 4. Effective Temperature:
See pages 57, 58, 59, 60 and 61 of the Heating, Ventilating, Air Conditioning Guide 1938.

SECTION III—Paragraph 2. Design Outside Temperature for Heating Load:
See page 138 of the Heating, Ventilating, Air Conditioning Guide 1938. Assume minus 15 degrees outside in calculating heating load.

SECTION III—Paragraph 3. Design Temperature for Cooling:
See page 148 of the Heating, Ventilating, Air Conditioning Guide 1938. Design temperatures for cooling shall be Outside 95 F Dry-bulb and 75 F Wet-bulb; Inside 80 F Dry-bulb and 67 F Wet-bulb; (50 per cent Relative Humidity). Outside 80 F Dry-bulb and 75 F Wet-bulb; Inside 78 F Dry-bulb and 65 F Wet-bulb; (52 per cent Relative Humidity).

SECTION V—Paragraph 1. Heat Transfer Rate for Building Construction:
See Chapter 5 of the Heating, Ventilating, Air Conditioning Guide 1938.

SECTION V—Paragraph 2. Heat and Moisture Emitted by Persons:
See Chapter 3, pages 73, 74, 75 and 76 of the Heating, Ventilating, Air Conditioning Guide 1938.

SECTION V—Paragraph 3. Heat Emission from Appliances:
See Chapter 8, page 160, of the Heating, Ventilating, Air Conditioning Guide 1938.

SECTION V—Paragraph 4. Solar Heat Entering Enclosure:
See pages 150, 151, 152, 153, 154, 155, 156 and 157 of Chapter 8 of the Heating, Ventilating, Air Conditioning Guide 1938.

SECTION V—Paragraph 5. Infiltration:
See Chapter 6, pages 121 to 131 of the Heating, Ventilating, Air Conditioning Guide 1938.

SECTION V—Paragraph 6. Ventilation:
The Code provides for a positive supply of outside air of not less than 10 cfm per stated number of occupants. However, where the City Ordinances require a greater volume of outside air than 10 cfm the City Ordinance must govern.

SECTION VII. Apparatus Rating:
Copies of the Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers, Standard Code for Testing and Rating Steam Unit Heaters, Standard Code for Testing Steam Unit Ventilators, Standard Code for Testing and Rating Air Cleaning Devices, may be obtained upon application to the Secretary of the American Society of Heating and Ventilating Engineers, 51 Madison Ave., New York, N. Y. Standard Method of Rating and Testing Air Conditioning Equipment and Standard Method of Rating and Testing Mechanical Condensing Units may be obtained by addressing the Secretary of the *American Society of Refrigerating Engineers*, 37 West 39th St., New York, N. Y.

SECTION VIII—Paragraph 1. City Water Temperature:
See Chapter 24, page 482, and Chapter 25, pages 508 and 509 of the Heating, Ventilating, Air Conditioning Guide 1938.

SECTION IX—Paragraph 2. Noise Control:
See Chapter 30 of the Heating, Ventilating, Air Conditioning Guide 1938 for Acceptable Noise Levels.

Local Requirements

Permits: An air conditioning system invariably contains the elements that bring these systems under the jurisdiction of five municipal departments of the City of Chicago. These departments are: (1) Board of Health. (2) Boiler Department. (3) Building Department. (4) Electrical Department. (5) Fire Prevention Bureau.

Other departments may also have jurisdiction. The contractor, before proceeding with the installation of an air conditioning system, should be properly protected by obtaining all of the required permits. When in doubt, information respecting legal requirements and permit fees may be obtained by consulting the various departments.

Pamphlets and Reprints Relating to City Requirements: The Chicago Municipal Sanitary Code of 1931 has not been reprinted; therefore, information required to conform with this Code can only be obtained by consulting the revised Chicago Municipal Code of 1931 or by applying to the Chicago Board of Health for such information.

The Ordinances regulating refrigeration have been mimeographed and a copy of same may be obtained on request from the Department for the Inspection of Steam Boilers, Unfired Pressure Vessels, and Cooling Plants. For other City regulations, the respective municipal department should be consulted.

Study of Summer Cooling in the Research Residence Using a Small Capacity Mechanical Condensing Unit

(Part 2)

By A. P. Kratz*, S. Konzo**, M. K. Fahnestock***, and E. L. Broderick† (*MEMBERS*)
Urbana, Ill.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the *National Warm Air Heating* and *Air Conditioning Association* and the University of Illinois.

Acknowledgments

The results presented in this paper were obtained in connection with the summer cooling investigation (1937) in the Research Residence‡ at the University of Illinois conducted by the Engineering Experiment Station of which M. L. Enger, dean of the College of Engineering, is the director, and in the Department of Mechanical Engineering of which O. A. Leutwiler, professor of Mechanical Engineering Design, is the head. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is also due to R. B. Engdahl, research graduate assistant, and to R. B. Knight, special research graduate assistant, for services rendered in connection with the investigation, and to the various manufacturers who cooperated by furnishing instruments and equipment.

Heat Absorbed by the Structure

It was observed in connection with Fig. 3 that during the period of continuous plant operation, when the actual heat absorbed by the cooling coil was only 65 per cent of the maximum calculated load on the Residence, the indoor dry-bulb temperature rose from 78.1 to 79.5 F, or a total rise of 1.4 F. During the season of 1936 a similar rise in indoor temperature was also observed. For example, on July 7, 1936[7] when the outdoor maximum temperature was 104.0 F, the calculated cooling load reached a maximum value of 37,000 Btu per hour, the actual heat absorbed by the cooling coil was 24,000 Btu per hour, and the indoor air temperature rose from 78.5 to 81.5 F, or a total rise of 3.0 F. In both examples, the cooling plant in use did not absorb heat at the rate indicated by the maximum calculated value of the cooling load. In view of the high rate at which the calculated cooling load increased as compared with the actual heat absorption, the magnitude of the rise in the indoor air temperature that was actually obtained appeared to be extremely small. The fact that this rise in the indoor air temperature was not greater has been attributed to heat absorbed by the building structure and furniture and not in turn given up to the indoor air.

The approximate amount of heat that would be absorbed by the Residence structure if the indoor air temperature was allowed to increase 1.0 F and equilibrium was assumed to be established under these new conditions was obtained as shown in Table 4 and Fig. 6. The weights of the different materials (columns 6 and 7) comprising the structure were calculated from the space occupied by each material, as determined by actual measurement, (columns 3 and 4) and the density of the material (column 5). The temperature rise (column 9) for those materials which were subjected to a 1.0 F rise in indoor air temperature on both sides of the material, such as a partition wall, was assumed to be also 1.0 F.

Owing to the existence of a temperature gradient, the temperature rise in those materials which were exposed to a 1.0 F rise in indoor air on one side and to a constant temperature on the other side, as in the case of an exposed wall, would be less than 1.0 F. The magnitudes of these temperature rises, based on the assumption that the indoor air temperature was allowed to increase 1.0 F and that equilibrium was established under the new conditions, were obtained by calculating the

*Research Professor, Engineering Experiment Station, University of Illinois.
**Special Research Assistant Professor, Engineering Experiment Station, University of Illinois.
***Research Assistant Professor, Engineering Experiment Station, University of Illinois.
†Research Assistant, Engineering Experiment Station, University of Illinois.

‡The Research Residence in Urbana, Illinois, was built, furnished, and completely equipped specifically for research work in warm air heating by the *National Warm Air Heating and Air Conditioning Association* in December, 1924.

[7]Loc. cit. Note 5.

Part 1 published in May, 1938.
For presentation at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938.

Table 4—Heat Absorbed by the Research Residence

Item No.	Item	Volume, Cu Ft		Density,[a] Lb per Cu Ft	Weight, Pounds		Spec. Heat[b] Btu per Lb per Deg	Temp. Rise in Material,[c] Deg F	Btu Absorbed per Deg F Rise in Indoor Air Temp.	
		1st St.	2nd St.		1st St.	2nd St.			1st Story	2nd Story
1	2	3	4	5	6	7	8	9	10	11
a	Plaster, partition wall	75.3	62.1	103.8	7820	6450	0.24	1.00	1878	1548
b	Plaster, exposed wall	37.9	61.0	103.8	3940	6340	0.24	0.80	756	1217
c	Plaster, ceiling	52.3	55.7	103.8	5430	5780	0.24	1.00 & 0.73	1303	1012
d	Glass, interior doors	0.3	.....	164.0	50	.....	0.20	1.00	10	.....
e	Glass, exposed	0.8	0.8	164.0	130	130	0.20	0.25	7	7
f	Studdings, partition wall	30.7	31.3	38.0	1170	1190	0.65	1.00	761	773
g	Studdings, exposed wall	29.9	41.8	38.0	1140	1590	0.65	0.64	474	661
h	Joists, ceiling	68.5	73.0	38.0	2600	2770	0.65	1.00 & 0.51	1690	918
i	Flooring	97.2	99.6	38.0	3690	3780	0.65	0.50 & 1.00	1200	2457
j	Wood, doors	20.4	15.2	28.0	570	430	0.65	1.00	370	280
k	Wood, door frames	51.8	45.9	57.0	2950	2620	0.65	1.00	1918	1702
l	Wood, window frames	36.0	35.5	38.0	1370	1350	0.65	1.00	890	878
m	Wood, baseboard	8.3	11.5	57.0	470	660	0.65	1.00	306	429
n	Wood, top moulding	5.9	1.2	57.0	340	70	0.65	1.00	221	46
o	Wood, mantel	2.0	.....	57.0	110	.....	0.65	1.00	72	.....
p	Wood, stairs	11.2	.....	38.0	430	.....	0.65	1.00	280	.....
q	Tile in bathroom	.....	4.9	115.0	.....	560	0.21	1.00	.....	118
r	Plasterboard wall	.....	4.3	68.0	.....	290	0.40	1.00	.....	116
s	Panelling	1.9	.....	38.0	70	.....	0.65	1.00	46	.....
t	Total	.....	.....	.....	.....	.....	.....	.....	12,182	12,162

[a]Values in column 5 for density obtained from Kent's *Mechanical Engineer's Handbook*, 10th edition, pp. 183-185.
[b]Values in column 8 for specific heats obtained from Marks' *Mechanical Engineer's Handbook*.
[c]Values in column 9 for temperature rise in material based on rise of 1.0 F in the indoor air temperature maintained until equilibrium conditions are attained.

temperature gradient through the wall, making use of the general equations of heat flow and the accepted values of the conductivities of the material. In the case of an exposed wall, calculations for the temperature gradient, and hence the surface temperatures, were based first on an indoor air temperature of 80 F and an outdoor air temperature of 95 F. The average of the two surface temperatures of the lath and plaster was then considered as the temperature of the slab comprised of the lath and plaster, and the average of the surface temperature of the sheathing and lap siding was considered as the temperature of the slab composed of these materials. Similar calculations were then made for an indoor air temperature of 81 F and an outdoor air temperature of 95 F. The difference in the temperatures of each slab of material obtained by these two calculations was considered as the temperature rise for that material. A graphical representation of the temperatures established in an exposed wall, for values of the indoor air temperature of 80 F and 81 F, is shown in Fig. 6. It may be noted that for an increase in the indoor air temperature of 1.0 F, the increases in temperature for the lath and plaster and for the studs were 0.80 F and 0.64 F respectively. The increases in temperature for the sheathing and lap siding were negligibly small. Similar calculated values for other materials are shown in column 9 of Table 4.

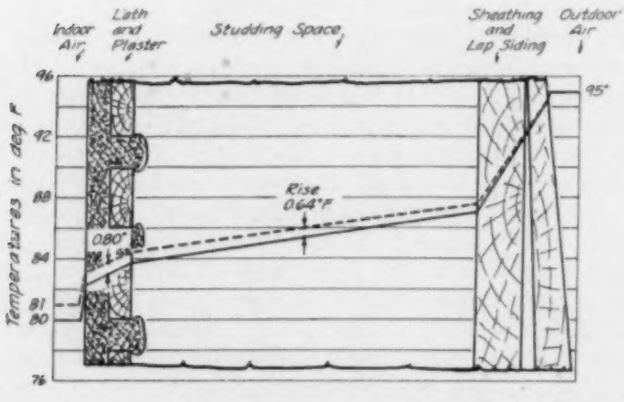


Fig. 6—Temperatures of materials comprising wall section. (Calculated values based on conditions of thermal equilibrium)

The total amounts shown in columns 10 and 11 in Table 4 indicate that the heat absorbed or given up by the Research Residence was 12,182 Btu on the first story and 12,162 Btu on the second story, or a total amount of 24,344 Btu for each one degree change in indoor air temperature. These values are for the building structure alone and do not include an allowance for the heat that could be absorbed by the furniture, floor coverings, light fixtures, books, and equipment. Therefore, an actual change of 1.0 F in the indoor air temperature is representative of a very large heat exchange. If the indoor air temperature were reduced 1.0 F a large amount of heat would be given up to the air by the structure and would represent a large additional load on the plant. Conversely, if the indoor air temperature rose 1.0 F the heat absorbed by the walls would tend to relieve the plant. For example, if the indoor air temperature changes 1.5 F during a four hour period, the rate at which heat is being absorbed or given up by the structure will be greater than 9,000 Btu per hour; which is equivalent to ¾ ton of refrigerating capacity during the four hour period.

These results emphasize the necessity, particularly where the plant capacity is limited, of starting the cooling plant each day before the indoor temperature has risen above that required for comfort conditions. If the indoor temperature is allowed to rise above 80 F, a comparatively high capacity will be required, even on a mild day, in order to effect a rapid decrease in the air temperatures, particularly at a time when the load on the cooling plant is increasing. Although these results apply to a larger extent to residence conditions, where the occupancy load is small, the effect of heat exchange between the air and the structure should also receive consideration in the case of other types of structures and under other operating conditions, particularly in installations in which a rapid reduction in the indoor air temperature is desired.

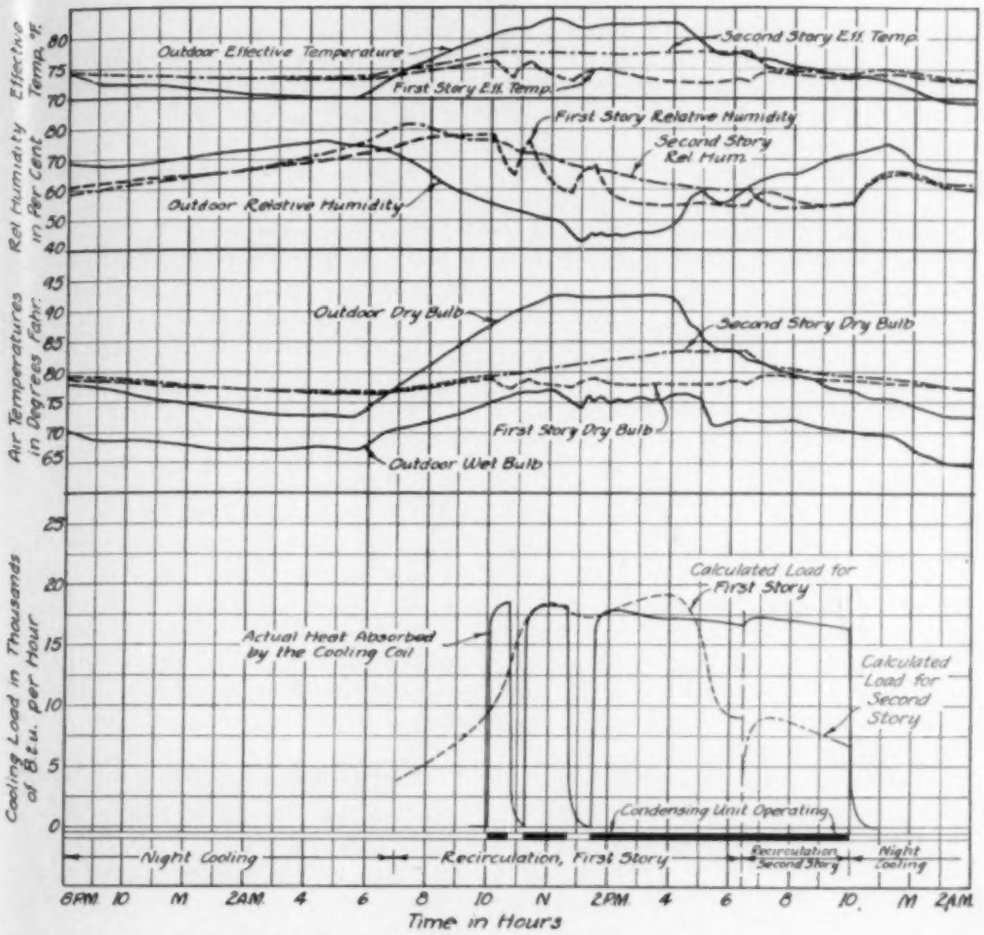

Effective Temp. °F.
Outdoor Effective Temperature
First Story Eff. Temp.
Second Story Eff. Temp.
Rel Humidity in Per Cent
Outdoor Relative Humidity
First Story Relative Humidity
Second Story Rel. Hum.
Air Temperatures in Degrees Fahr.
Outdoor Dry Bulb
Outdoor Wet Bulb
First Story Dry Bulb
Second Story Dry Bulb
Cooling Load in Thousands of B.t.u. per Hour
Actual Heat Absorbed by the Cooling Coil
Calculated Load for First Story
Calculated Load for Second Story
Condensing Unit Operating
Night Cooling
Recirculation, First Story
Recirculation, Second Story
Night Cooling
6PM 10 M 2AM 4 6 8 10 N 2PM 4 6 8 10 M 2AM
Time in Hours

Fig. 7—Actual and calculated cooling load on Residence and air temperatures. Test No. 19, Series 2-37, August 17, 1937

Results of Tests with Cooling on First Story During the Day and on Second Story at Night

General Conditions

The test made on August 17, 1937, was selected as typical of the operation obtained when cooled air was distributed to the first story rooms only until 6:30 p. m., and to the second story rooms only after 6:30 p. m. The results for the tests, made under conditions of operation for Series 2-37, are shown in the second and third columns of Table 1 and in Fig. 7.

The values listed in the second column of Table 1 show the average conditions that were maintained in the first story rooms at 3:30 p. m.; while the values listed in the third column show the average conditions that were maintained in the second story rooms at 7:30 p. m. The outdoor air temperature was about 93 F for a period of four hours and the day may be considered as a medium warm day.

Operating Characteristics

The operating characteristics for this method of operation are shown in Fig. 7. Outdoor air had been circulated through all the rooms of the house during the night previous to the test, from 8 p. m. to 7 a. m., by means of the basement fan. At 7 a. m. the recirculation of the indoor air, together with the addition of outdoor air for the purpose of ventilation, was started on the first story alone. Practically the same quantities of both recirculated air and ventilation air, as were circulated on both stories in the first series of tests, were circulated on each individual story in this second series of tests. Hence the quantities of air circulated, based on one story alone, were approximately equivalent to six recirculations per hour and two changes of outdoor air per hour, making a total of eight air changes per hour delivered by the fan. The ratio of the heat brought in by ventilation air to the total heat gain for the story was greater in this series than it was in the first series of tests.

On the test shown in Fig. 7 the condensing unit operated intermittently from 10:05 a. m. to 1:31 p. m. As in previous tests, the intermittent periods of plant operation were accompanied by large changes in the relative humidity and in the effective temperature. The comfort conditions during the off-periods of the plant were not as satisfactory as those maintained during the on-periods of the plant. These results are consistent with all of the results of tests made over the period of from 1934 to 1936, inasmuch as they all indicated that the indoor comfort conditions were more satisfactory on relatively hot days when the plant operated continuously than they were on the milder days when the plant operated intermittently. In this connection it may be mentioned that the use of a cooling unit or operating method which was just sufficient to maintain the desired indoor temperature at all times would eliminate these off-periods and the accompanying periods of high humidity. Any method of control or operation which would tend to modulate the capacity of the cooling unit to more nearly conform with the cooling requirements should prove more satisfactory than the intermittent operation of a unit having fixed capacity.

On this test, the condensing unit operated continuously from 1:31 p. m. to 10 p. m. The indoor comfort conditions on the first story remained constant and were entirely satisfactory until 6:30 p. m. From 6:30 p. m. until 10 p. m., when the cooled air was distributed to the second story rooms only, the indoor temperature, humidity, and effective temperature on the first story increased slightly, but the conditions were acceptable. It is probable that on extremely severe days, the conditions on the first story after 6:30 p. m. might prove uncom-

fortable, and an additional investigation under these severe weather conditions is required.

It may be observed that the temperatures in the rooms on the second story increased steadily from 76.8 F at 7 a. m. to 83.5 F at 6:30 p. m. Since no air was recirculated on the second story during this interval, and the windows remained closed, the air conditions in the second story rooms were not satisfactory. The contrast between the warm, stuffy environment on the second story and the cool, fresh environment on the first story was particularly striking. The conditions were so noticeably different, particularly in late afternoon, that the occupants of the house preferred to remain on the first story. This limitation and restriction in the usage and occupancy of the entire house may prove to be undesirable in some installations, where sewing rooms and utility rooms are located on the second story. The normal living habits of the occupants should, therefore, be considered in the application of zoned installations of this nature.

During the interval when cooling was applied only on the first story, the relative humidity on the second story decreased from 77 per cent at 10 a. m. to 60 per cent at 6:30 p. m., whereas ordinarily when no cooling was done in the house the indoor relative humidity tended to remain fairly constant. This reduction in relative humidity on the second story was caused by a reduction in the moisture content of the air and not merely by the higher air temperature. The moisture content decreased from 116 grains per pound of dry air at 10 a. m. to 104 grains per pound at 6:30 p. m. This reduction of 12 grains per pound of dry air on the second story accompanied a reduction in the moisture content on the first story amounting to 37 grains per pound, and indicated that the circulation of relatively cool dry air on the first story appreciably affected the conditions on the second story. As a result of this the effective temperature on the second story remained practically constant in spite of the fact that the dry-bulb temperature increased.

At 6:30 p. m. when the cooled air was distributed only through the registers on the second story, the reduction in relative humidity and temperature occurred quite rapidly during the first hour. The reduction in temperature occurred very slowly after that time due to the effect of the heat absorbed in the structure. This latter effect is also indicated, as shown in the bottom part of Fig. 7, by the wide discrepancy between the calculated cooling load on the second story and the actual load absorbed by the plant. During the interval from 8 p. m. to 10 p. m. the actual load was more than twice as great as the calculated load, while the indoor air temperature on the second story decreased from 80.5 F to 79.3 F, giving a total drop of only 1.2 F. The degree of comfort on the second story, as measured by the effective temperature, did not attain that represented by 74.5 deg until approximately two hours after the cooling was begun on the second story. Even at 10 p. m., 3½ hours after the cooling was begun on the second story, the effective temperature was 74 deg, only slightly below the upper border line of comfort. It is most probable that on a hotter day the conditions on the second story would not be completely satisfactory between the hours of 6:30 p. m. and 10 p. m. Additional data on this point are desirable.

In general, therefore, subject to the limitations discussed, the method of cooling on the first story during the day and on the second story during the night may be considered as reasonably satisfactory. The method requires a zoned duct system in which all of the rooms on one story are supplied with air from a trunk duct separate from those supplying air to other stories. Such an arrangement may not be practical in all installations. The restriction in the usage of the house may prove objectionable in some cases. Furthermore, the same unit when used to cool the entire house during the comparatively mild season of 1937 proved almost as satisfactory without restricting the space suitable for occupancy. On the whole, the comfort conditions attained on the first story rooms were slightly more satisfactory than those maintained when the small condensing unit was used to cool the entire house. However, the comfort conditions maintained during the off-periods of the plant were subject to improvement, and furthermore more satisfactory conditions as a whole would have resulted if the number of off-periods of the plant could have been reduced. The conditions attained on the second story were just on the upper border line of comfort, and a larger capacity would have been required in order to produce a more rapid reduction in the temperature.

Summary of Seasonal Results

Table 5 gives a summary of the total quantities obtained during the summer of 1937. This season was comparatively cool and in this respect was in decided contrast to the preceding hot summer. The total of 862 degree-hours above 85 F, which was about one-fifth that obtained for 1936, is a good indication of the difference between the two seasons. The total numbers of days with temperatures 85 F or over and 90 F or over for the season of 1937 were 61 and 23 respectively. The

Table 5—Summary of Results of Tests for Season of 1937

Weather Data		
1.	Total hours above 85 F for season of 1937	301.9
2.	Total hours above 90 F for season of 1937	40.1
3.	Total degree-hours above 85 F for season of 1937	862.4
4.	Total degree-hours above 90 F for season of 1937	63.7
Night Air Cooling		
5.	Total number of nights with cooling by fan	76
6.	Total running time for fan, hours	845.9
7.	Average rate of power input to fan, watts	453
8.	Total power input to fan, kilowatt-hours	390.2
Air Recirculation Without Cooling		
9.	Total number of days	75
10.	Total running time for fan, hours	674.3
11.	Average rate of power input to fan, watts	432
12.	Total power input to fan, kilowatt-hours	291.4
Air Recirculation with Cooling		
13.	Total number of tests	28
14.	Total running time for fan, hours	220.6
15.	Average rate of power input to fan, watts	437
16.	Total power input to fan, kilowatt-hours	97.4
Compressor Operation		
17.	Total number of hours	199.0
18.	Average rate of power input, watts	1396
19.	Total power input, kilowatt-hours	277.9
20.	Total quantity of cooling water, gallons	11,961
Combined Totals		
21.	Total running time for fan including night air cooling, hours	1756.8
22.	Total power input to fan including night air cooling, kilowatt hours	779.0
23.	Total power input to compressor and fan including night air cooling, kilowatt-hours	1056.9

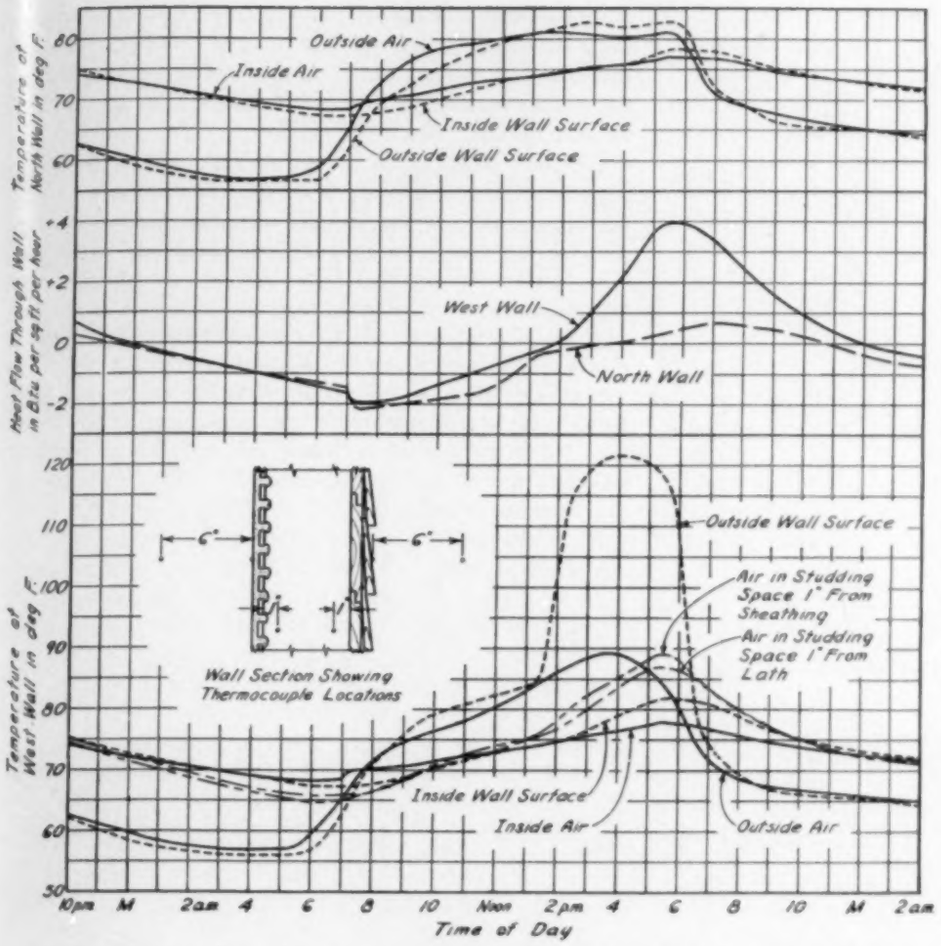


Fig. 8—Heat flow through walls and temperatures at wall sections for August 13, 1937

severity of the season, as measured by degree-hours, is apparently determined largely by the total number of days on which the maximum temperature exceeds 90 F. The outdoor humidities were more nearly normal during 1937, and were much higher than those obtained during the hot season of 1936.

From Item (21) in Table 5 it may be observed that the total time that the fan was running, including cooling with outdoor air at night and recirculation during the day, was 1756.8 hours. The total cost for electricity for operating the fan, based on a rate of 3.1 cents per kilowatt-hour for electricity, was $24.13. The total time of operation of the compressor was 199.0 hours and the cost of electricity for this purpose was $8.61. The total amount of water actually used for the condensing unit was 11,961 gal. At the prevailing rate of 33 cents per thousand gallons, the total cost for water was $3.95. The cost for both electricity and water for the season was $36.69. Even allowing for two days in early June during which the compressor was not ready for operation, the total seasonal cost would not have exceeded $40. It may be noted that the cost of operation of the condensing unit was only one third of the total cost, the cost of operation of the circulating fan accounting for the other two thirds. Since the variations in seasonal demand were extremely wide, a comparison of the cost obtained with the small condensing unit with that obtained with other types of cooling units might to a certain extent be misleading.

Results of Study with Heat-Flow Meters

Temperature Studies

The data obtained from the two heat-flow meter installations over a period of 29 days were plotted on a continuous chart, and this chart was used as a basis for the analysis of results. A typical section from the chart for a 28 hour period is shown in Fig. 8. It may be noted that the temperatures were continually changing and that thermal equilibrium was never attained. This cyclical variation both in the temperatures and in the direction of heat flow does not conform with the assumption of thermal equilibrium, which is invariably made in the calculation of heat losses or gains under given conditions of sun effect and temperature difference between the indoors and outdoors. The top set of curves shows the air and surface temperatures obtained at the north wall section, while the lower set shows those obtained at the west wall section. The two curves in Fig. 8 between the upper and lower sets show the actual amounts of heat passing through the inside surface of the two walls, as indicated by the readings of the heat-flow meters. These meters were located one on each wall, directly below the thermocouples installed to obtain the temperature gradient through the wall.

It may be observed from Fig. 8 that there existed an appreciable interval between the time at which the outdoor air temperature attained a maximum value and that at which the indoor air temperature reached a maximum. In order to obtain the average values of the times at which various parts of the wall sections attained their maximum temperatures the data obtained from the 29 days were averaged, and are shown in Fig. 9. The values for the west wall, which was exposed to the sun, were not appreciably different from those for the north wall, which was not exposed to the sun. The lag between the times at which the outdoor air temperature was a maximum and the indoor air temperature was a maximum was approximately 2.5 hours. Also, the lag between the times at which the outdoor surface temperature reached a maximum and indoor surface temperature attained a maximum was approximately two hours.

Studies of Heat Transmission

Preliminary calculations of the conductance of the wall section, as determined from actual measurements of

the surface temperatures and the heat flow through the heat-flow meters, gave results that were materially less than the values obtained for similar walls from laboratory determinations under conditions of thermal equilibrium. This indicated either that the conductance value determined from laboratory investigations was in error or that conditions of thermal equilibrium were not attained in the actual wall exposed to normal temperature variations. The former supposition is not tenable,

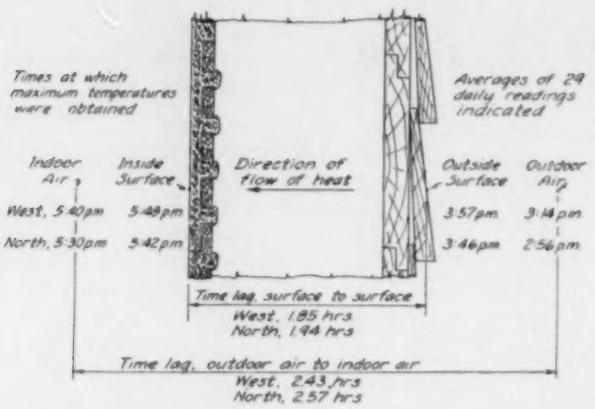


Fig. 9—Average values of times at which maximum temperatures were attained in wall sections

whereas the latter condition was actually known to exist.

This discrepancy between the measured and the calculated values of the heat flow through the wall is illustrated by the curves in Fig. 10. The calculated values were determined from the following equation[8]:

$$H = \frac{Ue\ I}{f_o} + U\ (t_o - t_i)$$

in which

H = heat transmitted through wall in Btu per square foot per hour,
U = overall coefficient of heat transfer, Btu per square foot per hour per degree difference in temperature,
e = coefficient of absorption of solar radiation, (0.40 for weathered white paint),
I = solar intensity normal to the surface, in Btu per square foot per hour,
f_o = outside surface coefficient, Btu per square foot per hour per degree Fahrenheit,
t_o = outdoor air temperature, degrees Fahrenheit,
t_i = indoor air temperature, degrees Fahrenheit.

It may be noted from Fig. 10 that the maximum value of the calculated heat flow through the west wall was equal to 6.3 Btu per square foot per hour and occurred at 4 p. m.; whereas the maximum value of the heat actually transmitted into the room as measured by the heat-flow meter, was 3.9 Btu per square foot per hour and occurred at 6 p. m. The time lag between these maximum values was 2 hours.

It was also observed in these studies that at certain times of the day the temperature in the studding space was higher than those of both the indoor and outdoor surfaces of the wall. That is, under these conditions the heat flow in the wall was occurring simultaneously to the outdoors and to the indoors. The calculated values of heat transmission are of little significance under such conditions; the significant values being given by the heat-flow meter, which on account of its location on the inside surface of the wall measures the actual flow of heat into the room.

During periods when the outdoor air temperature was increasing, the temperature of the material comprising the wall structure was usually also increasing. As explained in the section on heat absorbed by the structure, the total amount of heat impressed on the wall appears not only as heat passing through the heat-flow meter but also as heat which tends to increase the temperature of the wall structure, and the discrepancy between the maximum values shown in Fig. 10 has been attributed to this factor, although it is also possible that the presence of the heat meter itself may have a disturbing effect. It is evident, however, that in walls of the type tested, the lack of equilibrium conditions results in changes in the amount of heat absorbed or given up by the structure and affects the amount of heat transmitted through the wall; so that the calculated values of the heat flowing through the wall at any given time, when used as an index of the heat actually transmitted to the air in the structure, are not only meaningless, but may also be misleading. Furthermore, under summer conditions the maximum amount of heat transmitted through the inside surface of the wall into the inside of the structure may be less than the maximum heat flow through the wall as calculated from the indoor and outdoor conditions.

Summary and Conclusions

The following summary and conclusions may be considered as applying to the Research Residence and the conditions under which the tests were conducted.

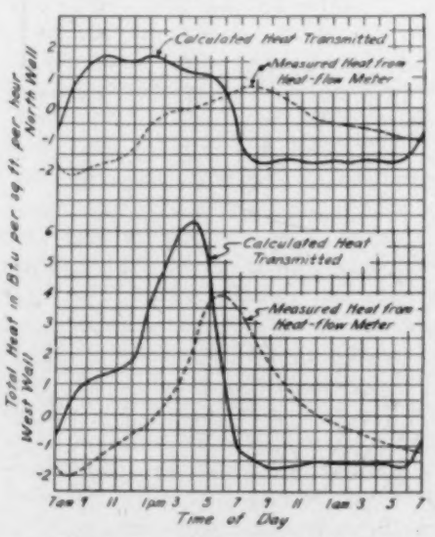


Fig. 10—Heat flow through walls of Research Residence for August 13, 1937

(1) In order to maintain the same effective temperature, the unit having the smaller capacity was operated to maintain an indoor dry-bulb temperature somewhat lower than that maintained with a unit having a larger capacity. Furthermore, it was advisable to allow for a rise in the indoor temperature on hot days when the cooling plant did not have sufficient capacity to maintain a constant temperature.

(2) The cooling capacity of 17,300 Btu per hour was found to be sufficient for both stories for days on which the maximum outdoor temperature rose to 95 F and the minimum outdoor temperature during the night preceding the test did not exceed 71 F. The daily median temperature (average of maximum and minimum outdoor temperature) of 83 F may be considered as

[8]Loc. cit. Note 6. Appendix B, Method Used for Calculation of Cooling Load, pp. 131-134.

the upper limit of the range of application of the small-sized condensing unit.

(3) A statistical study of the daily median temperatures during a given number of seasons was made, and from it the limitations of the application of the small condensing unit were determined.

(4) Subject to limitations over a small portion of the cooling season, the use of a condensing unit of small capacity was shown to be feasible for residences, especially in those cases in which the initial cost of installation is a more important factor than the maintenance of ideal indoor conditions.

(5) The fact that the rise in indoor air temperature accompanying the operation of the small condensing unit on a warm day was small was attributed to the heat absorbed by the building structure and furniture.

(6) The calculations for the heat absorbed by the structure of the Research Residence indicated that a rise of 1.0 F in the indoor air temperature was equivalent to a heat absorption of over 24,000 Btu.

(7) A comparatively large plant would probably be required, even on a mild day, in order to effect a rapid decrease in the indoor air temperatures, particularly if the load on the cooling plant were increasing.

(8) The off-periods occurring during periods of intermittent operation of the plant were accompanied by comparatively large changes in the relative humidity and in the effective temperature.

(9) Subject to the limitation of restricted occupancy, the method of cooling on the first story during the day and on the second story during the night may be considered as reasonably satisfactory.

(10) The studies made with heat-flow meters applied to the walls indicated that considerable discrepancy existed between the measured values and the calculated values of the heat flow through the wall. Furthermore, thermal equilibrium was never attained in the wall section under actual conditions.

(11) Except under equilibrium conditions, the total amount of heat impressed on the wall appears not only as heat transmitted to the indoor air but also as heat serving to increase the temperature of the wall structure. The discrepancy between the maximum amount of heat actually transmitted to the indoor air, as measured by the heat-flow meter, and the maximum calculated heat transmission for the wall has been attributed to heat absorbed by the wall structure.

NEWS OF LOCAL CHAPTERS

Coils for Cooling and Dehumidification Subject of Minnesota Meeting

April 11, 1938. Approximately 85 members and guests of Minnesota Chapter met at the Minnesota Union and were called to order by Pres. R. E. Backstrom. The minutes of the previous meeting were approved and the treasurer's report was given.

Secy. F. C. Winterer read a letter from Manitoba Chapter inviting Minnesota Chapter members to a meeting to be held in Winnipeg on May 28. It was voted to accept the invitation and it was suggested that President Backstrom appoint a Transportation Committee.

H. M. Betts reported for the Membership Committee and plans for the May 16 party were outlined by M. H. Bjerken and F. W. Legler.

The Nominating Committee's report was read by N. D. Adams, chairman, and there being no further nominations from the floor, it was moved by Prof. F. B. Rowley and seconded by G. A. Dahlstrom that nominations be closed and the report of the Nominating Committee be accepted. The motion was carried and the nominations were as follows:

President—J. E. Swenson
Vice-President—F. C. Winterer
Secretary-Treasurer—M. H. Bjerken
Board of Governors—F. H. Schernbeck, R. E. Backstrom.

Motion was made by Mr. Legler, seconded by F. F. Lange, and carried that a committee be appointed to select an air conditioned meeting place for the 1938-39 meetings.

President Backstrom introduced the speaker of the evening, William Goodman, Trane Co., LaCrosse, Wis., who spoke on the Use of Coils for Cooling and Dehumidification. His talk covered the relation of the surface to dry- and wet-bulb temperature of the air and the dehumidification factors entering into the design of the coils. Mr. Goodman also spoke on the necessity for considering the relation of the location of direct expansion coils to condensing units and the proper selection of expansion valves under various conditions of installation. Mr. Goodman's talk proved interesting and of considerable value to the members and guests, and before adjournment he answered numerous questions.

A.S.H.V.E. Officers and Council Members Guests of Western New York Chapter

April 25, 1938. At the April meeting held at the University Club, Buffalo, the Western New York Chapter was honored in having as its guests all of the officers of the A.S.H.V.E., as well as those members of the Council who had attended a meeting of the Council earlier in the day. An attendance was recorded of 59 for dinner and 80 for the session which followed.

Nominations for Chapter officers for the year 1938-39 as prepared by the Board of Governors were presented to the meeting, and Secy. W. R. Heath was instructed to send ballots to the members eligible to vote.

All other business was dispensed with for the evening and Pres. B. C. Candee turned the meeting over to L. A. Harding, who acted as toastmaster and introduced the guests.

E. Holt Gurney, Toronto, Ont., president of the A.S.H.V.E., spoke on the subject of Air Conditioning Brings Human Problems to the Engineer, predicting a promising future for air conditioning and presenting aspects of air conditioning problems not usually considered.

J. F. McIntire, Detroit, 1st vice-president of the Society, gave an address on The Engineer, mentioning especially the resourcefulness required and using as an illustration the unique method of preventing land slides by freezing the edge of excavations.

F. E. Giesecke, College Station, Tex., 2nd vice-president of the A.S.H.V.E., described activities of the Texas Chapter and told of the vast amount of air conditioning being installed in that section of the country, especially in some of the more simplified forms such as attic ventilating systems.

A. J. Offner, New York, treasurer of the Society, told briefly of the finances of the A.S.H.V.E.

Publication of the HEATING, VENTILATING, AIR CONDITIONING GUIDE was discussed by Secy. A. V. Hutchinson.

W. L. Fleisher, New York, chairman of the Committee on Research, told briefly of some of his experiences while visiting England and France. He showed how much more advanced air conditioning is in the United States than in Europe and also described some of the problems which foreign countries have

in providing air conditioning in many underground gas-proof and bomb-proof shelters being constructed.

Prof. A. P. Kratz, Urbana, Ill., spoke of some of the research work now being conducted.

C. W. Farrar explained to his fellow members that the location of Hot Springs, Va. was only 400 miles from Buffalo and he recommended that members of the Western New York Chapter attend the Semi-Annual Meeting of the Society, June 20-22.

President Candee thanked Mr. Harding for the capable and entertaining manner in which he conducted the meeting, and expressed the Chapter's appreciation to the Society's Officers and Council for providing a very entertaining and profitable evening.

Rockefeller Center Air Conditioning Described to Massachusetts Chapter

April 12, 1938. Sixty members and guests attended the meeting of Massachusetts Chapter, which was held at the Massachusetts Institute of Technology, Cambridge, Mass., and preceded by supper at Walker Memorial Cafeteria.

The principal speaker was C. W. Walton, Jr., mechanical engineer, Rockefeller Center, Inc., New York, who gave an interesting description of this group of buildings and described the air conditioning systems and various problems in connection with their operation. At the close of his talk, Mr. Walton answered a number of pertinent questions asked by several members present.

A talking motion picture, entitled Heat and Its Control, produced by Johns-Manville, was shown by Mr. Anderson. This proved interesting to the members present, especially from the point of view of insulation, according to the report of Secy. H. C. Moore.

Illinois Chapter Enjoys Talk on Forced Air Heating

April 11, 1938. Pres. S. I. Rottmayer called the meeting of the Illinois Chapter to order in the Brevoort Hotel, Chicago, with 135 members and guests present. The minutes of the March meeting were read by Secy. C. E. Price, after which E. M. Mittendorff, chairman of the Membership Committee, introduced the following new Chapter members: O. J. Greenwood, W. D. Fleak, W. A. Stahl, and G. B. Tupper.

R. E. Hattis reported on the status of the Illinois Engineering Council with the information that a constitution for the proposed organization was being drafted, and that interested societies would be given an opportunity to review the constitution before it was adopted.

Tom Brown, chairman of the Meetings Committee, announced that the annual meeting in May was scheduled for the Bismarck Hotel and that Pres. E. Holt Gurney, Toronto, and first vice-president J. F. McIntire, Detroit, had been invited.

G. F. Nightingale, a former resident of Chicago who was present as a visitor, was introduced by President Rottmayer.

The speaker of the evening was S. Konzo, University of Illinois, Urbana, Ill. He was introduced by J. H. Van Alsburg and his subject was Trends in Forced Air Heating. Professor Konzo presented the findings of research being conducted at the Research Residence in Urbana and particularly stressed the following items which have been indicated by tests:

(1) Fuel consumptions were decreased with a given furnace installation when the rate of oil input was decreased.

(2) The fuel consumption for a furnace designed to burn oil (having 100 sq ft of heating surface) was materially less than that for a conversion furnace (having 70 sq ft of heating surface).

(3) Although the maintenance of relatively high percentages of CO_2 in the flue gas is advisable, the most effective means of providing for minimum oil consumption is to operate with low flue gas temperatures.

(4) The comfort conditions in the room were improved, particularly during zero weather, with more nearly continuous operation of the fan, such as can be provided by a two-volume delivery fan.

(5) Very marked improvements in the floor temperature were obtained by converting the plain perforated grille type baseboard register to the deflecting type baseboard register.

(6) The extremely rapid developments and refinements in the field have lead towards compact units that are economical in operation and that provide for optimum comfort under all conditions of service.

Pres. Gurney Visits Michigan Chapter

April 28, 1938. The regular meeting of the Michigan Chapter was held at the Detroit Leland Hotel at 6:30 p. m., with approximately 60 members and guests in attendance.

Pres. F. J. Feely introduced six past presidents of the Chapter who were present. He then asked for a report of the Secretary, G. H. Tuttle, and the Program Committee chairman, E. E. Dubry. Mr. Dubry announced that the next meeting would be held at the Birmingham Golf Club.

It was announced by President Feely that the following three members would serve as the Nominating Committee for 1939: R. K. Milward, chairman, G. D. Winans, and W. G. Boales.

E. M. Harrigan, past president of the Chapter, was then introduced and served as toastmaster during the evening.

E. Holt Gurney, Toronto, Ont., president of the A.S.H.V.E., was the guest speaker and discussed Society activities and some of the problems confronting the air conditioning engineer.

After Mr. Gurney's address, Mr. Harrigan introduced J. H. Walker, member of the A.S.H.V.E. Council and vice-chairman of the Committee on Research, who discussed some of the work of the committee and the Research Laboratory at Pittsburgh.

The meeting adjourned at 10:00 p. m., according to the report of Secretary Tuttle.

Radiant Heating, Motors and Controls Subjects of Cleveland Meetings

April 5, 1938. Northern Ohio Chapter members and guests met at the Cleveland Club, Cleveland, with 17 present for dinner and 60 attending the meeting.

The application for membership of J. E. Lamping was submitted and accepted.

Pres. Philip Cohen called for a vote on the motion to send L. T. Avery to the Semi-Annual Meeting of the Society at Hot Springs, Va. The motion was carried and the treasurer was authorized to pay Mr. Avery's round-trip railroad fare.

The president then called for Secy. C. A. McKeeman to read a motion which was presented in a letter by Prof. G. L. Tuve, chairman of the Standards Committee, recommending the acceptance of the Cleveland Standards of Air Conditioning for Human Comfort, which were prepared by *Air Conditioning Association, Inc.*, with certain deletions, changes, etc. Considerable discussion followed and a motion was made and carried: That the standards of air conditioning as recommended by the Standards Committee in their letter be adopted as the code of the Northern Ohio Chapter. Secretary McKeeman was instructed to advise the *Air Conditioning Association, Inc.*, and the *Cleveland Association of Consulting Engineers* of this action.

A motion was made and passed that all present affiliate members, together with those accepted between now and October 1, 1938, shall expire October 1, 1939. Also that all affiliate members accepted after October 1, 1938 shall expire one year from date of acceptance.

A discussion on air conditioning was then lead by Mr. Hansen.

D. J. Purinton, Mahoney-Troast Construction Co., New York, N. Y., was the speaker of the evening and was introduced by President Cohen. Mr. Purinton's talk on Radiant Heating was

favorably received by the Chapter, and his subject and delivery held the interest of those present.

March 14, 1938. The Northern Ohio Chapter meeting at the Cleveland Club, Cleveland, was attended by 83 members and guests, 34 being present for dinner. Following the reading and acceptance of the minutes, reports were received from the Program, Publications, and Legislation Committees.

Prof. G. L. Tuve reported for the Committee on Standards to the effect that the Committee had reviewed the Cleveland Standards of Air Conditioning for Human Comfort. Some minor corrections were recommended; otherwise the Committee's report was favorable.

L. T. Avery discussed the 44th Annual Meeting of the A.S.H.V.E., giving details of the meeting of Chapter delegates, as well as an explanation of the functioning of the Nominating Committee. Since the actual nominations are made at the Society's Semi-Annual Meeting, a motion was made to send Mr. Avery to Hot Springs, Va., in June.

Professor Tuve discussed the different grades of membership in the Chapter, since the national organization recognizes as Chapter members only those who are also members of the A.S.H.V.E. The opinion was expressed that emphasis should be placed on national membership and the question was referred to the officers and Board of Governors.

The speakers of the evening were then introduced and Secy. C. A. McKeeman reports that Messrs. Mobarry and Russell gave interesting talks on Motors and Motor Controls.

Pres. Gurney Addresses Manitoba Chapter

April 12, 1938. Manitoba Chapter held a meeting at the Fort Garry Hotel, Winnipeg, at 6:30 p. m. with 22 members and guests present and Pres. D. F. Michie presiding. Roll call was taken and the minutes of the March meeting were read and adopted.

The report of the Nominating Committee as follows was approved and the slate was recommended for election at the annual meeting of the Chapter on May 28:

President—William Worton
Vice-President—William Glass
Secretary—E. J. Argue
Treasurer—H. R. Eade
Board of Directors—D. F. Michie, P. L. Charles, Frank Thompson

The guest speaker of the evening was E. Holt Gurney, Toronto, president of the A.S.H.V.E., who was introduced by J. E. Yates, Sr. Mr. Gurney gave a very interesting address on the activities of the Society, which was followed by adjournment at 9:00 p. m.

Iowa-Nebraska Chapter Participates in Conference

April 8, 1938. The eighth meeting of the Iowa-Nebraska Chapter was held at Iowa State College, Ames, Iowa. This meeting was conducted in conjunction with the Third Annual Heating, Ventilating and Air Conditioning Conference at the College, and the program for April 8 was sponsored jointly by the Iowa-Nebraska Chapter of the Society and the Iowa State College.

The program consisted of technical lectures by men prominent in the industry. A. L. Walters, president of the Chapter, acted as chairman during the morning session, and N. B. Delavan, chairman of the Program Committee, presided during the afternoon. Many of those in attendance inspected the new air conditioning test equipment recently set up.

The conference was terminated by a dinner held at the Sheldon-Munn Hotel, when T. R. Agg, dean of engineering, Iowa State College, was toastmaster. President Walters gave a short history of the Society and explained the activities of the local Chapter since its organization. Following Mr. Walters, Prof. R. A. Norman, chairman of the Nominating Committee, reported.

Prof. M. P. Cleghorn, head of the mechanical engineering department, briefly outlined the research facilities of the college. Professor Cleghorn explained that their purpose was to conduct research and experimental work relative to the development of heating, ventilating and air conditioning plants. He told of the desire of the College to cooperate with those identified with the field of air conditioning, and cited the benefits to be derived from cooperative effort.

The final speaker of the evening, Herman Seid, patent and engineering counsel, Auditorium Conditioning Corp., New York, N. Y., gave a forceful and effective presentation of the subject, Air Conditioning Patents and Their Effect on the Industry's Development, which was well received according to the report of Secy. W. R. White.

Pneumatic Temperature Control Systems Subject of Washington, D. C. Meeting

April 13, 1938. The regular meeting of the Washington, D. C. Chapter was held at Wesley Hall, with a technical session preceded by dinner and approximately 35 members and guests present.

The business meeting was called to order at 8:00 p. m. by Pres. L. S. Ourusoff, who announced that the May meeting was scheduled to be held in the Federal Reserve Building with C. S. Leopold, Philadelphia, as speaker.

The attention of Chapter members was called to the forum arranged by the Department of Health and those interested were urged to attend.

The next order of business was the report of the Nominating Committee which presented the following slate for 1938-39:

President—S. P. Eagleton.
Vice-President—S. L. Gregg.
Secretary—E. V. Fineran.
Treasurer—W. R. Lockhart.
Board of Governors—L. S. Ourusoff, Glegge Thomas, T. H. Urdahl.

There were no further nominations from the floor and arrangements were made for sending ballots to the Chapter membership.

President Ourusoff introduced John Meyers, Johnson Service Co., who spoke on Pneumatic Temperature Control Systems. His talk was accompanied by lantern slides covering principles of pneumatic control and illustrating the various types of control equipment essential for this work.

Secy. L. F. Nordine advises that an approximate balance of $76.00 was reported by the Treasurer.

Philadelphia Chapter Elects Officers and Hears Discussion by T. H. Urdahl

April 14, 1938. Philadelphia Chapter members and guests met at the Engineers Club and the minutes of the previous meeting were read and approved. Edwin Elliot reported that the Auditing Committee had found the Treasurer's accounts in perfect order.

H. H. Mather, chairman of the Membership Committee, stated that 51 applications for membership in the Chapter had been obtained during the past year.

Plans for the May meeting were outlined by Vice-Pres. H. H. Erickson.

Secy. C. B. Eastman read the slate of the Nominating Committee for officers of the Chapter for the coming year. Pres. L. P. Hynes called for further nominations from the floor and as none were offered, a vote was taken instructing the Secretary to cast a ballot for the election of the following officers for the coming season:

President—H. H. Erickson
Vice-President—R. F. Hunger
Secretary—H. H. Mather
Treasurer—C. B. Eastman
Board of Governors—H. F. Rettew, W. A. Bornemann, L. P. Hynes.

Following the election, Mr. Hynes presented a new gavel to the incoming president, Mr. Erickson, stating that he desired

to cherish the gavel which he had used throughout the year as a memento of his service as president of the Philadelphia Chapter.

M. F. Blankin was called upon by President Erickson to present the gift of a camping kit to Mr. Hynes in recognition of his efficient administration.

The speaker of the evening was T. H. Urdahl, consulting engineer, Washington, D. C., who discussed the subject of Air Distribution Problems in Air Conditioning.

Atlanta Chapter Is Host to F. E. Giesecke

April 20, 1938. A special meeting of Atlanta Chapter was held at Georgia School of Technology in honor of Prof. F. E. Giesecke, College Station, Tex., 2nd vice-president of the A.S.H.V.E.

Professor Giesecke was introduced by Pres. E. W. Klein and delivered an address on the Influence of Building Construction on Air Conditioning. Treating his subject in an interesting and instructive manner, Professor Giesecke presented a wealth of valuable data on heat gain and heat loss in buildings, together with many other timely facts having to do with heating and cooling. An interesting summary of the early history of the Society was also presented and in addition, Professor Giesecke told of the research program now being carried on.

Members of other engineering societies had been invited to attend the lecture and many responded.

Secy. C. T. Baker announced that after the May meeting, the regular sessions would be resumed in the Fall.

Air Conditioning with Lithium Chloride Subject of Cincinnati Chapter

April 11, 1938. Members and guests of Cincinnati Chapter met at the Engineers Club with 35 in attendance. Preceding the meeting, dinner was enjoyed at the Old Vienna Restaurant, with 25 members present and guests including Mr. and Mrs. F. R. Bichowsky.

Pres. I. B. Helburn called the session to order and announced that the next meeting would be held at the Hartwell Country Club, when a short movie on steel making was to be shown by Tom Byrd.

Mr. Bichowsky, Dow Chemical Co., Midland, Mich., was introduced by President Helburn and addressed the Chapter on the subject of Comfort and Industrial Air Conditioning with Lithium Chloride. With the aid of lantern slides showing curves and graphs, and illustrations of apparatus, the speaker explained the effectiveness of various absorbents. He stated that there are four classes of materials to remove water from air as follows: (1) Solid absorbents; (2) pure liquids; (3) solids which combine chemically with water; and (4) solids which combine with water from the air and form a liquid such as calcium chloride and lithium chloride. The apparatus commonly used for passing air through lithium chloride solution is a string type tower, having vertical cotton strings, in which the lithium chloride flows down the strings, while the air is passing between the strings at about 500 ft velocity.

After the address, Mr. Bichowsky answered many questions, which were instructive, and an enthusiastic vote of thanks was given to the speaker for his able and interesting presentation.

The meeting adjourned at 10:15 p. m. according to the report of Secy. H. E. Sproull.

Electrostatic Dust Precipitator Interests Pittsburgh Chapter

April 11, 1938. The Pittsburgh Chapter meeting in the private dining room of Stouffer's Restaurant was called to order by Pres. J. F. Collins, Jr., and was attended by 65 members and guests. The minutes of the March meeting were read by Secy. T. F. Rockwell and were approved.

The following committee chairmen reported on the activities of their respective committees: E. C. Smyers—Program Committee; P. A. Edwards—Membership Committee; R. A. Miller—Convention Committee; F. C. McIntosh—Research Committee; and F. B. Mahon—Air Conditioning Code.

The technical session was opened with short talks by two senior students from the Department of Mechanical Engineering of Carnegie Institute of Technology. Daniel Supon spoke on Dehumidification by Silica Gel. P. A. Schwartz followed with a brief discussion of Comfort Cooling by Mechanical Refrigeration.

The principal speaker of the evening was G. W. Penney, research engineer, Westinghouse Electric & Manufacturing Co. Mr. Penney had charge of the development of the electrostatic dust precipitator for the cleaning of air in ventilation work and chose this device as the subject of his paper. He began by pointing out the difference between this recently developed device and the earlier Cottrel precipitator, showing that in the new model the dust particle is first ionized in one chamber and then precipitated in a second chamber. The speaker exhibited a number of slides showing recent installations, while he reviewed some operating experiences. There was also a small working model available, which Mr. Penney demonstrated by using tobacco smoke. The many questions which followed indicated the appreciation of the audience for the paper.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 37 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by June 15, 1938, th ese candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ADLER, HERMAN, Student, New York Univ., New York, N. Y.	Harold Connell P. J. Barbieri	Edwin Sternberg M. C. Giannini
BACHMAN, A. J., Student, Pratt Inst., Brooklyn, N. Y.	C. R. Hiers J. W. Hunter (*A.S.M.E.*)	H. S. Cameron (*A.S.M.E.*) F. S. Egilsrud (*A.S.M.E.*)
BACHOFER, H. A., JR., Mgr. Htg. & Air Cond. Dept., Mid-West Plbg. Co., Salina, Kans.	W. F. Ryan E. A. Jones	K. T. Davis K. M. Davis (*Non-Member*)
BYSOM, L. L., Assoc. Mech. Engr., Puget Sound Navy Yard, Bremerton, Wash. (*Reinstatement*)	F. J. Pratt M. N. Musgrave	C. F. Twist C. G. Zokelt
CAMPBELL, E. K., JR., Secy., E. K. Campbell Htg. Co., Kansas City, Mo. (*Advancement*).	E. K. Campbell, Sr. N. W. Downes	A. H. Sluss H. J. Nottberg
CASKEY, L. H., JR., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.	C. M. Humphreys T. F. Rockwell	P. A. Edwards F. C. McIntosh
COST, G. W., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.	C. M. Humphreys T. F. Rockwell	F. C. McIntosh P. A. Edwards
COWELL, R. J., Consulting Engr., New York, N. Y. (*Reinstatement*)	L. P. Hynes H. L. Alt	W. J. Searle, Jr. L. A. Childs
DAUMAN, ARNOLD, Student, New York Univ., New York, N. Y.	J. F. Kern, Jr. S. S. Dauman (*I.E.S.*)	C. C. Clark (*I.E.S.*) D. B. Lucas (*A.S.A.E.*)
DAVIS, W. P., Air Cond. Advisor, Buffalo Niagara Elec. Corp., Buffalo, N. Y.	W. E. Voisinet J. J. Yager	S. W. Strouse J. K. Hawk
DELANY, J. V., Student, Worcester Polytech. Inst., Worcester, Mass.	R. P. Kolb (*A.S.M.E.*) D. G. Downing (*A.S.M.E.*)	F. W. Roys (*A.S.M.E.*) M. L. Price (*A.S.M.E.*)
DE WILDE, M. P., Air Cond. Engr., Carrier Engrg. S. A., Ltd., Johannesburg, South Africa	E. G. Carrier Jacobus Ehlers	C. A. von Christierson D. C. Lindsay (*S.C.M.E. of S.A.*)
DRESCHER, F. E., Sales Engr., Straus-Frank Co., Houston, Texas	R. F. Taylor R. E. Chrone	C. A. McKinney R. J. Salinger
ENDERS, C. E., Factory Repr. & Engr., Electrol, Inc., Portland, Ore.	E. C. Willey J. D. Kroeker	T. E. Taylor R. D. Watt
FARNES, B. W., Sales Mgr., Control Equipment Co., Portland, Ore.	J. D. Kroeker T. E. Taylor	E. C. Willey S. H. Graf (*A.S.M.E.*)
FORDERBRUGGEN, K. J., Engr., Mankato Natural Gas Co., Mankato, Minn.	T. W. Russell H. M. Betts	J. C. Davidson C. E. Lund
GAYMAN, P. D., Br. Mgr., Johnson Service Co., Cleveland, Ohio	F. C. McIntosh M. F. Rather	L. T. Avery J. P. Jones
GRABMAN, H. B., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.	C. M. Humphreys T. F. Rockwell	P. A. Edwards F. C. McIntosh
GROSVOLD, F. E., Htg. & Vtg. Contractor, F. E. Grosvold, Eau Claire, Wis. (*Reinstatement*).	E. N. Bowles R. C. Lockhart (*Non-Member*)	F. E. Gerrish (*Non-Member*) J. C. Ryan, Jr. (*Non-Member*)
HOEY, J. K., Pres. & Mgr., Crater Metal & Engrg., Inc., Medford, Ore.	J. D. Kroeker T. E. Taylor	E. C. Willey R. S. Daniels (*Non-Member*)
JEHLE, FERDINAND, Director, Research Lab., Hoffman Specialty Co., Stamford, Conn. (*Advancement*)	G. L. Tuve A. P. Kratz	W. H. Severns L. P. Saunders
JENKINS, F. H., Student, Worcester Polytechnic Inst., Worcester, Mass.	F. W. Roys (*A.S.M.E.*) D. G. Downing (*A.S.M.E.*)	A. J. Staples (*S.P.E.E.*) R. P. Kolb (A.S.M.E.)
KADEL, G. B., Design Engr., E. R. Squibb & Sons, Brooklyn, N. Y.	E. J. Ritchie R. W. Cumming	Adolf Winterroll (*N.Y.P.E.*) F. C. Chase (*N.Y.P.E.*)
KINCAIDE, M. C., Chief Air Cond. Engr., Timken Silent Automatic Co., Detroit, Mich. (*Advancement*)	F. R. Bishop G. H. Tuttle	W. G. Boales G. D. Winans
LANCE, J. F., Supt., Harrigan & Reid Co., Detroit, Mich. (*Reinstatement*)	W. J. Whelan E. M. Harrigan	R. K. Milward E. H. Clark
LEVY, M. I., Pres., Viking Air Cond. Corp., Cleveland, Ohio (*Advancement*)	H. W. Fiedler J. G. Foster	J. A. Heller B. H. Broh (*A.S.M.E.*)
MATZ, G. N., Mech. Engr., A. Ernest Dambly, Philadelphia, Pa.	A. E. Dambly M. F. Blankin	C. B. Eastman L. C. Davidson
MCKINLEY, JOHN, Owner, McKinley Ref. Supply Co., Fort Worth, Texas	J. H. Van Alsburg H. W. Skinner	R. K. Werner C. O. Towne
MCKINNEY, W. J., Dist. Mgr., American Blower Corp., Atlanta, Ga. (*Advancement*)	F. R. Dickenson H. E. Barth	A. P. Darlington H. E. Paetz
MILLER, A. T., Insulation Sales, The Barrett Co., New York, N. Y.	F. B. Rowley A. B. Algren	R. E. Backstrom P. W. Gumaer
MITCHELL, JACK, Mgr. Air Cond. Dept., Straus-Frank Co., Houston, Texas (*Reinstatement*)	R. F. Taylor R. E. Chrone	R. J. Salinger C. A. McKinney
NEWMAN, H. E., Asst. Mgr., B. A. Newman Co., Fresno, Calif.	N. H. Peterson W. W. Cockins	H. V. Hickman C. E. Bentley
POLLARD, M. J., Installation Engr., Sidles Co., Airtemp Div., Omaha, Nebr.	F. E. Prawl M. J. Stevenson	M. M. Shapiro L. L. Daubert
SHEPPERD, P. D., Sales Engr., Johnson Service Co., San Francisco, Calif.	F. W. Hook C. E. Bentley	G. M. Simonson James Gayner
SMITH, N. J., Engr., Frigidaire Corp., Dayton, Ohio	C. E. Lewis H. B. Hull	C. F. Wood C. F. Henney (*A.S.R.E.*)
STE-MARIE, G. P., Examiner, Dept. of Labour, Montreal, Que., Can. (*Reinstatement*)	A. J. Dickey M. W. Shears	G. L. Wiggs Leo Garneau
ZITZMAN, F. T., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.	C. M. Humphreys T. F. Rockwell	F. C. McIntosh P. A. Edwards